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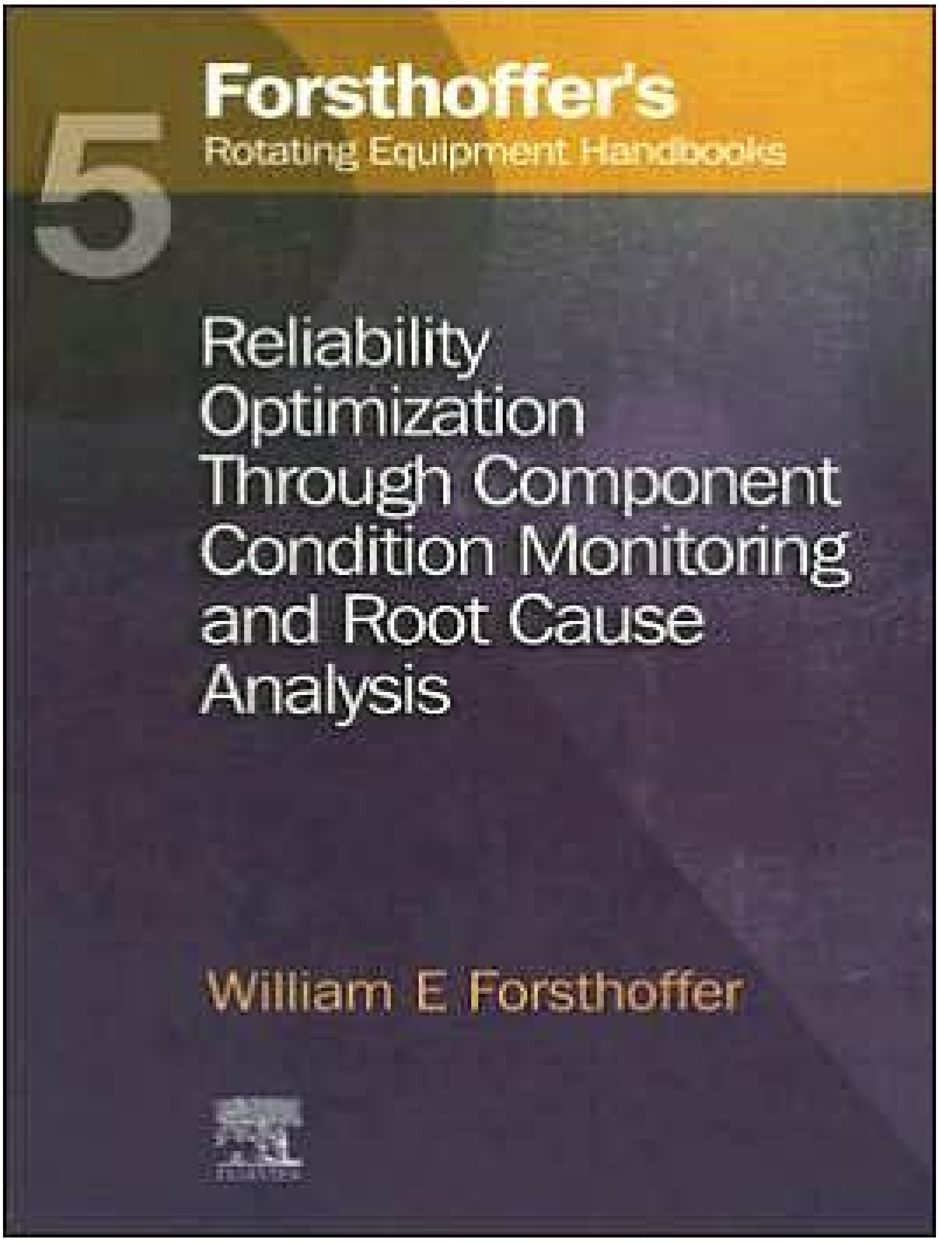
Forsthoffer's Rotating Equipment Handbooks

Vol 5: Reliability Optimization through Component Condition and Root Cause Analysis



5

Forsthoffer's

Rotating Equipment Handbooks

Reliability
Optimization
Through Component
Condition Monitoring
and Root Cause
Analysis

William E Forsthoffer

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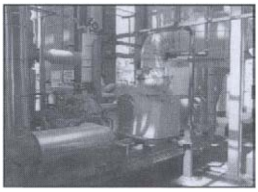


Preface

This Series has evolved from my personal experience over the last 40 years with the design, selection, testing, start-up and condition monitoring of Rotating Equipment. Most of the concept figures were originally written on a blackboard or whiteboard during a training session and on a spare piece of paper or I beam during a start-up or a problem solving plant visit.

My entire career has been devoted to this interesting and important field. Then and now more than ever, the cost of rotating equipment downtime can severely limit revenue and profits. A large process unit today can produce daily revenues in excess of 5 million US dollars. And yet, the Operators, Millwrights and Engineers responsible for the safety and reliability of this equipment have not been afforded the opportunity to learn the design basis for this equipment in practical terms. I have also observed in the last ten years or so, that the number of experienced personnel in this field is diminishing rapidly.

Therefore the series objective is to present, in User friendly (easy to access), practical terms (using familiar analogies), the key facts concerning rotating equipment design basis, operation, maintenance, installation and condition monitoring to enable the reader (Engineer, Operator and Millwright) to:

- Understand the effect of process and environmental changes on equipment operation, maintenance and reliability
- Condition Monitor equipment on a component basis to optimize up-time, mean time between failure (MTBF) and mean time to repair (MTTR)
- Select, audit and test the Equipment that will produce highest safety and reliability in the field for the lowest life cycle cost.

The hope is that the knowledge contained in this series will enable

Plant Operations, Maintenance and Engineering Personnel to easily access the material that will allow them to present their recommendations to management to solve existing costly problems and produce new projects of optimum reliability.

This volume, Reliability Optimization thru Component Condition Monitoring and Root Cause Analysis, details the effective method of component condition monitoring for use as both a predictive maintenance and root cause analysis tool. It also details the major failure causes, the authors proven root cause analysis procedure with exercises and case histories, installation, pre-commissioning planning, functional testing and commissioning, preventive maintenance strategies and more.

Acknowledgements

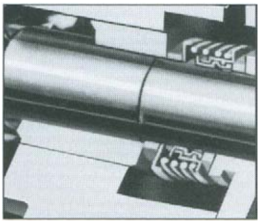
This series is a result of interactions with literally thousands of dedicated engineers, machinists, operators, vendors, contractors and students who have been an integral part of my career.

I consider myself very fortunate to have been associated with the best of the best mentors, business associates and dear friends throughout my career. Most especially, in chronological order Dick Salzmann, Bob Aimone, Merle Crane, Walt Neibel, the late Murray Rost, Mike Sweeney and Jimmy Trice. Bob, Merle, Murray and Mike have contributed specifically to the material in this series while Dick, Walt and Jimmy have tactfully kept me on track when necessary.

Special thanks to all of the global machinery vendors who have allowed me to use their material for my training manuals and now this publication.

Last but certainly not least; my career would not have been possible without the support, encouragement and assistance from my wife Doris and our children Jennifer, Brian, Eric, Michael and Dara.

A special additional note of thanks to Michael who helped assemble the material, and hopefully learned some in doing so, since he has elected to pursue a career in rotating machinery.



About the author

Bill Forsthoffer began his life-time career in rotating machinery in 1962 with De Laval Turbine Inc. as a summer trainee. After obtaining a Bachelor of Arts degree in Mathematics and Bachelor of Science degree in Mechanical Engineering, during which time he worked for De Laval part time in the Test, Compressor and Steam Turbine Departments, he joined De Laval full time in the Compressor Engineering Department in 1968. He was responsible for all phases of centrifugal compressor component and auxiliary design and also made many site visits to provide field engineering assistance for start up and problem resolution.

Bill joined Mobil Oil Corporate Engineering in 1974 and was responsible for all aspects of rotating equipment specification, technical procurement, design audits, test, field construction, commissioning, start-up and troubleshooting.

After 15 years at Mobil, Bill founded his own consulting company in 1990 and has provided rotating equipment consulting services to over 100 companies. Services include: project reliability assurance, training (over 7,000 people trained) and troubleshooting.

Bill is active in the industry as President of Forsthoffer Associates Inc., frequently writes articles for Turbo Machinery International Magazine and conducts many site specific and public workshops each year.

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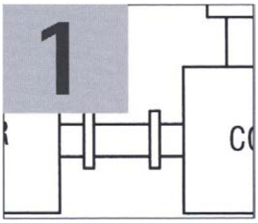


1

Reliability overview

- Introduction
- The end user's objectives
- Reliability terms and definitions
- Optimizing reliability

Introduction

Reliability optimization is an important part of plant revenue and profit. The objective of this volume is to provide information that will enable the reader to optimize reliability by implementing proven methods I have used throughout my career. The major components of reliability improvement are shown in Figure 1.1.

Volume objective

'To optimize site rotating equipment availability by implementation of practical'

- Site reliability audits
- Assessment methods
- Availability improvement plans
- Condition monitoring techniques
- Preventive and predictive maintenance plans
- Troubleshooting techniques

Figure 1.1 Volume objective

Before these objectives can be met and implemented, a number of important concepts and terms need to be reviewed and presented.

The end user's objectives

The objectives of the end user are shown in Figure 1.2.

The objectives

- Maximum reliability (on stream time)
- Maximum product throughput
- Minimum operating costs

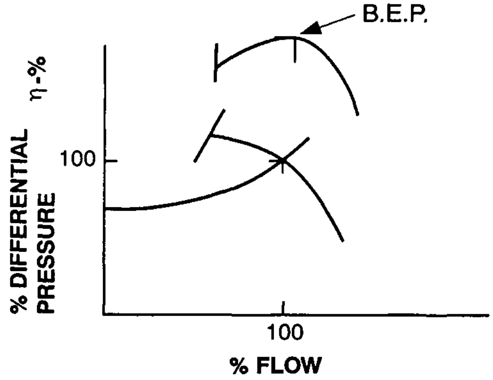


Figure 1.2 The objectives

In order to maximize profit, a piece of machinery must have maximum reliability, maximum product throughput and minimum operating cost (maximum efficiency). In order to achieve these objectives, the end user must play a significant part in the project during the specification and design phase and not only after the installation of the equipment in the field. Effective field maintenance starts with the specification phase of a project. Inadequate specifications in terms of instrumentation and the location of instrumentation will impact equipment reliability.

It is important to understand that the life span of rotating equipment is extremely long compared to the specification, design and installation phase. Refer to Figure 1.3.

A typical installation will have a specification, design and installation

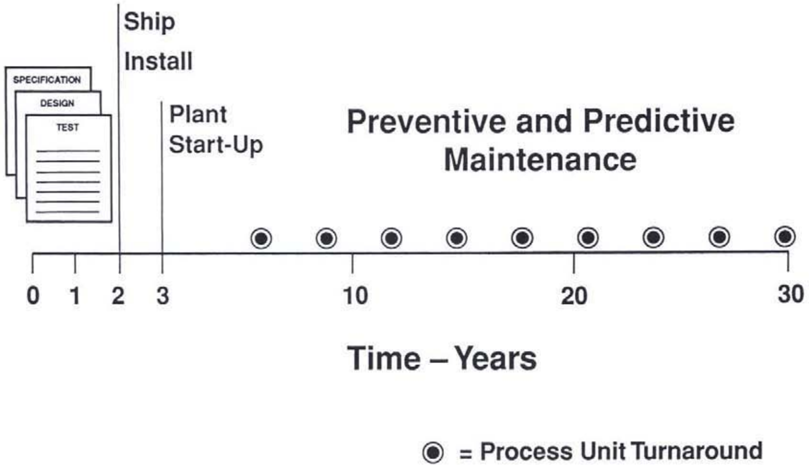

SPECIFICATION
DESIGN
TEST
Ship
Install
Plant
Start-Up
Preventive and Predictive
Maintenance
0
1
2
3
10
20
30
Time – Years
◉ = Process Unit Turnaround

Figure 1.3 The life span of rotating equipment

phase of only approximately 10% of the total life of the process unit. Improper specification, design or installation will significantly impact the maintenance requirements, maintenance cost and availability of a particular piece of machinery. Proper screening of equipment design (pre-bid technical meetings etc.) prior to equipment vendor selection establishes the foundation on which reliability is built. Likewise, enforcing shipment, construction, installation and commissioning specifications optimizes reliability and truly makes it 'cost effective' in terms of the life cycle of the equipment.

Reliability terms and definitions

Before we can optimize reliability, certain terms and definitions need to be presented. These terms are shown in Figure 1.4.

Reliability terms

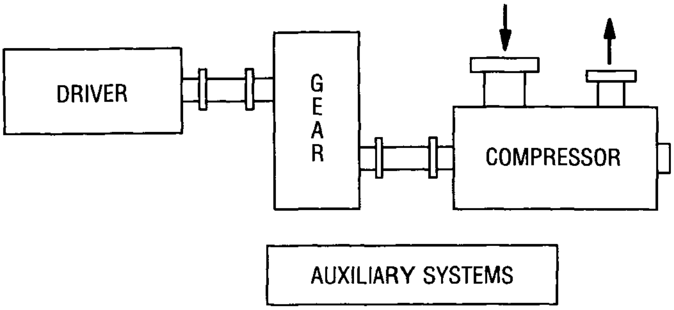
- Reliability
- Availability
- Maintainability
- Cost of unavailability

Figure 1.4 Reliability terms

Reliability

Reliability is the ability of the equipment unit to perform its stated duty without a forced (unscheduled) outage in a given period of time (see Figure 1.5).

The rotating equipment unit



Every piece of rotating equipment is part of a unit which consists of:

- The driver
- The driven
- The transmission devices
- The auxiliary systems

Figure 1.5 The rotating equipment unit

The definition of reliability for critical (unspared) equipment is presented in Figure 1.6.

Reliability – critical equipment

The amount of time equipment operates in one year

$$\text{Reliability (\% =)} \left(\frac{\text{Operating hours per year}}{\text{8760 hours}}\right) 100$$

Figure 1.6 Reliability – critical equipment

In the case of general purpose equipment (spared), reliability is not usually calculated since a spare unit should be available for operation if required. In the case of unreliable general purpose units, reliability could be defined as shown in Figure 1.7.

Reliability – general purpose (spared) equipment

Hours per year spared equipment operated as a percentage of the hours it was required to operate

$$\text{Reliability (\% =)} \left(\frac{\text{Yearly hours in operation}}{\text{Yearly main unit forced outage hours}} \right) 100$$

Figure 1.7 Reliability – general purpose (spared) equipment

Note in Figure 1.6 and 1.7 that reliability does not account for planned downtime for preventive and/or predictive maintenance.

Availability

Availability considers preventive and predictive maintenance downtime as shown in Figure 1.8.

Availability

The amount of time equipment operates in one year as a percentage of the available hours per year

$$\text{Availability (\% =)} \left(\frac{\text{Yearly operating hours}}{8760 - \text{planned downtime}} \right) 100$$

(T & Is or turnarounds)

Figure 1.8 Availability

One measure of both reliability and availability is mean time between failure or MTBF. See Figure 1.9.

Mean time between failure

$$\text{MTBF} = \frac{\text{Total operating hours}}{\text{Number of failures}}$$

Figure 1.9 Mean time between failure

Maintainability

Simply stated, maintainability is the ability to perform all maintenance activities; preventive, predictive and forced outage in a minimum time that requires rotating equipment unit shutdown. It is understood that the total maintenance time required will restore the unit to its original 'new' condition.

One parameter that can be used to measure maintainability is mean time to repair – MTTR as shown in Figure 1.10. The lower the MTTR, the greater the maintainability.

Mean time to repair

$$\text{MTTR} = \frac{\text{Total repair hours*}}{\text{Number of repair events}}$$

* includes corrective maintenance time

(actual labor hours as a result of design modifications)

Figure 1.10 Mean time to repair

Cost of unavailability

All terms discussed so far, reliability, availability and maintainability directly affect the product revenue of the plant. Product revenue is the value obtained from one days production expressed in local currency. At this point, note the amount of daily revenue from a process unit in your plant in Figure 1.11. Note that typical amounts can vary from $100,000 to over $5,000,000.00 per day depending on the process and the size of the unit.

Daily product revenue for a process unit

Date: ______________

Amount: ______________

Process unit type: ______________

Figure 1.11 Daily product revenue for a process unit

If a critical equipment unit suffers a forced outage or is out of service due to poor maintainability (extended repair time), the product revenue shown in Figure 1.11 will be lost for each day the critical equipment unit remains out of service.

Therefore, the cost of unavailability is the total of the values shown in Figure 1.12.

The cost of unavailability critical rotating equipment (per year)

- Lost product revenue x days forced outage
- Maintenance costs
- Replacement part cost
- Labor cost
- Unnecessary turnaround time*

* Assumes process unit start-up is delayed by activity

Figure 1.12 The cost of unavailability critical rotating equipment (per year)

The cost of unavailability can be a powerful tool to use in preparing reliability improvement plans.

Optimizing reliability

The key to reliability improvement is to build a solid program foundation. Figure 1.13 shows the reliability pyramid.

The success or failure of any reliability improvement program directly depends on obtaining and maintaining management support. Figure 1.14 presents guidelines for meeting this important objective.

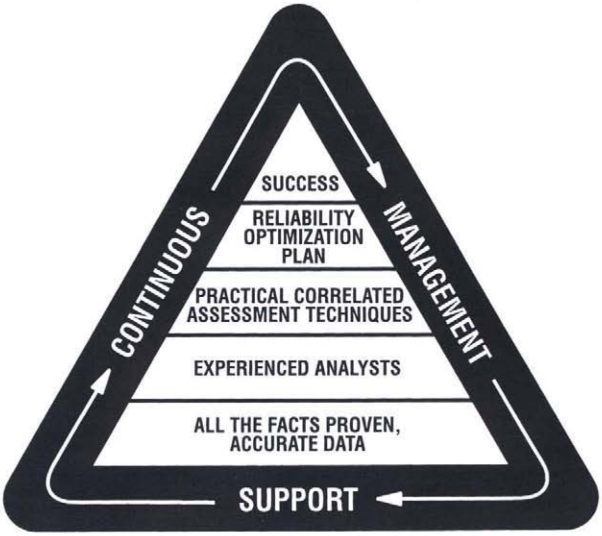

SUCCESS
RELIABILITY OPTIMIZATION PLAN
PRACTICAL CORRELATED ASSESSMENT TECHNIQUES
EXPERIENCED ANALYSTS
ALL THE FACTS PROVEN, ACCURATE DATA
CONTINUOUS
MANAGEMENT
SUPPORT

Figure 1.13 The Reliability Pyramid

Obtain and maintain management support by...

1. Clearly stating impact of problem on plant profit (cost of unavailability)
2. Prepare a brief statement of:
   - Problem
   - Impact on plant
   - Reliability improvement plan
3. Be confident!
4. Be professional
5. Be autonomous (do not expect management to do your job!!)
6. Provide timely updates

Figure 1.14 Obtain and maintain management support by...

Input data

Once management support is obtained, input data forms the foundation of the program. Figure 1.15 presents important guidelines concerning input data.

Reliability input data

- Include all the facts (operation, reliability, maintenance failure analysis, etc.)
  - Consider the machinery environment
  - Consider the entire system
- Only use proven data (don't guess!)
- Accuracy is most important – confirm data is correct

Figure 1.15 Reliability input data

The environment or surroundings for any piece of rotating equipment play an important part in determining the availability of that particular item (refer to Figure 1.16).

The rotating equipment environment

- Process condition change
- Piping and foundation change
- 'Unit' (driven, driver, transmission, auxiliaries)
- Ambient conditions

Figure 1.16 The rotating equipment environment

This figure shows that the rotating equipment environment is the process unit in which the equipment is installed. If any of these items are not taken into account, the accuracy of the conclusions reached during the assessment phase will be significantly reduced.

In my experience, most failures in predictive maintenance and troubleshooting exercises occur because the entire system in which the component operates is not considered. Every component in every piece of machinery operates in a system. Defining the system and all of the components contained therein is a very important step in successful problem analysis. Refer to Figure 1.17.

The concept of a system

- Think system!
- Every component is a part of a system
- In order to determine root causes, systems and not just components must be considered

Figure 1.17 The concept of a system

Experience counts!

Having experienced analysts to determine root causes of low reliability is the next step in building a strong program. Figure 1.18 suggests ways to build and develop a practical, strong analyst group.

Analyst development strategy

- Select experienced rotating equipment personnel
- Ideally, design and field experience are the best combination
- Provide site specific training
- Measure results
- Provide opportunities for networking with other specialists within and outside the company
  - User groups
  - Industry conferences
  - Regional conferences
- Include analysts in all phases of new projects

Figure 1.18 Analyst development strategy

Utilize practical, correlated assessment techniques whenever possible

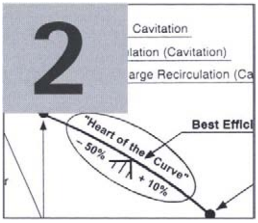
Today, many statistical methods are available to the analyst to determine causes of failure and to predict equipment and component life. The personal computer makes the use of these methods quick and easy.

However, the reader is cautioned to regard all statistical methods as only a part of the process. Whenever possible, actual data concerning failure rates should be used and the correlation of statistical methods should be defined. It should always be remembered that the basis for most statistical methods have evolved from industries where 'production components' are used, i.e. the electronics, automotive industries, etc. However, the rotating equipment unit regardless of type always becomes customized by virtue of its environment. That is, each rotating equipment unit has its own signature. Consequently, care must be exercised when applying statistical methods to rotating equipment reliability assessment. Figure 1.19 presents this important fact.

Statistical methods and rotating equipment

Since every rotating equipment unit, regardless of size represents a 'customized system' care must be exercised when assessing the results obtained by statistical methods.

Figure 1.19 Statistical methods and rotating equipment



2

The major causes of machinery failure

- Rotating equipment does not fail randomly
- The major causes of machinery failure – failure classifications
- Summary

Rotating equipment does not fail randomly

Regardless of the location, rotating equipment usually fails when we don't want it to ... on the weekend! In the Middle East it fails late on Wednesday afternoon. In other places, failure occurs late Friday afternoon! Are these events random failures? Can we predict them?

There is always a root cause of failure and there are indications in the failed component condition. However, general purpose equipment, because it is not usually continuously monitored (directly in the control room), certainly can appear to fail randomly.

Please refer to Figure 2.1.

Equipment does not fail randomly!

- There are root causes
- The condition of the failed component will change.

Figure 2.1 Equipment does not fail randomly!

Consider your plant's Bad Actor List. Has progress been made in reducing this list? Yes, it has! However, we frequently observe that once the root cause of the failure has been determined for a 'Bad Actor', it will eventually fail again. Why? It is because the process variables (parameters) affecting the failed component condition are not being monitored. How can we minimize random failures and our 'Bad Actor List'? By being aware of the major reasons for failure and by observing the condition of the machinery components.

Please refer to Figure 2.2.

How to stop ... firefighting (random failure)

- Know why failures occur (5 Why's)
- Condition monitor the major components
- Make it a 'team effort'

Figure 2.2 How to stop ... firefighting (random failure)

Will this involve more data collection, more work? Many times, workload and meetings are reduced.

It all comes down to ... Awareness, knowing what to look for.

In the following sections of this chapter, the root causes of machinery failures will be discussed in detail. In the next chapter, the ways to prevent machinery failures will be discussed.

The major causes of machinery failure – failure classifications

The causes of machinery failure can be grouped into the failure classifications noted in Figure 2.3. Note that usually, failures are the result of more than one cause.

Failure classifications

- Process condition changes
- Assembly/installation
- Operating procedures
- Design deficiencies
- Component wearout

Figure 2.3 Failure classifications

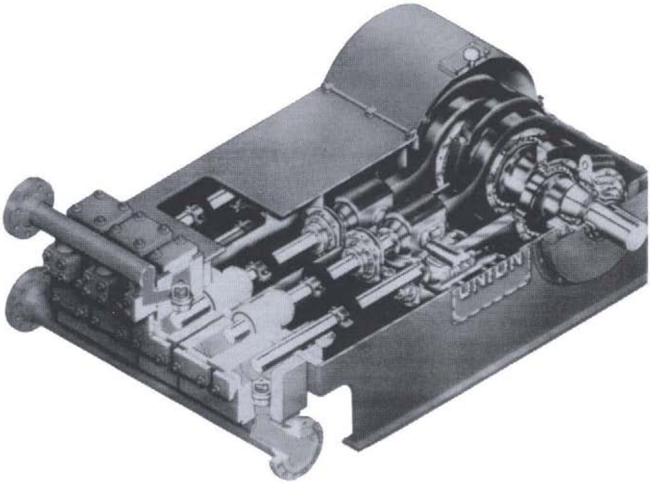


Figure 2.4 Positive displacement plunger pump

1. Process condition changes

This classification is the most overlooked in terms of troubleshooting. For this discussion, the most common type of driven equipment – pumps will be used.

There are two (2) major classifications of pumps, positive displacement and kinetic, centrifugal types being the most common. A positive displacement pump is shown in Figure 2.4. A centrifugal pump is shown in Figure 2.5.

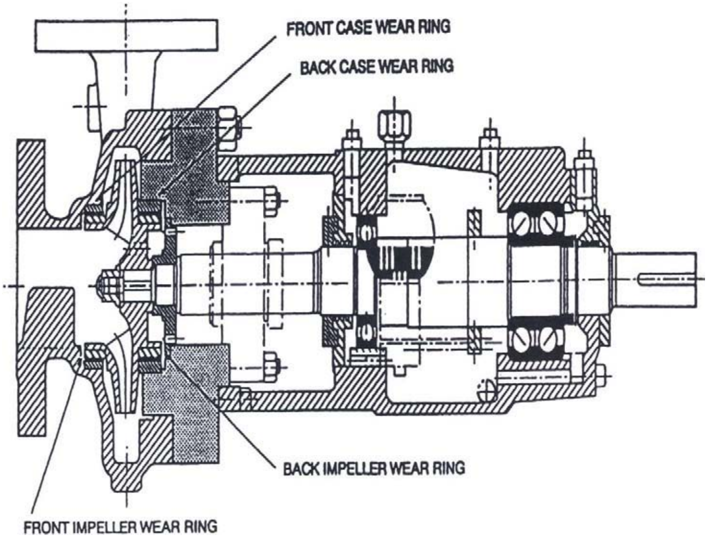

FRONT CASE WEAR RING
BACK CASE WEAR RING
BACK IMPELLER WEAR RING
FRONT IMPELLER WEAR RING

Figure 2.5 Centrifugal pump

In a typical refinery, greater than 95% of the installed pumps are the centrifugal type.

Positive displacement pumps increase the pressure of the liquid by operating on a fixed volume in a fixed space. The most common types of positive displacement pumps are listed in Figure 2.6.

Types of positive displacement pumps

Pulsating – non-continuous flow
- Plunger
- Diaphragm
- Piston

Rotary – continuous flow
- Screw
- Gear

Figure 2.6 Types of positive displacement pumps

The characteristics of positive displacement pumps are detailed in Figure 2.7.

Positive displacement pump characteristics

- Constant flow
- Variable pressure produced
- Require a pressure limiting device (PSV)
- Flow does not vary with specific gravity changes

Figure 2.7 Positive displacement pump characteristics

It is most important to remember that all driven equipment (pumps, compressors, fans, etc.) react to the process system requirements. They do only what the process requires. This fact is noted in Figure 2.8 for pumps.

Pump performance

- Pumps produce the pressure required by the process
- The flow rate for the required pressure is dependent on the pump's characteristics

Figure 2.8 Pump performance

Based on the characteristics of positive displacement pumps noted in Figure 2.7, positive displacement pump flow rate is not significantly affected by the process system. This fact is shown in Figure 2.9.

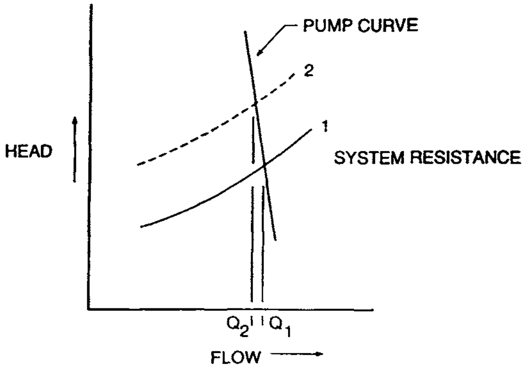


Figure 2.9 A positive displacement pump in a process system

Therefore, since the flow rate of a positive displacement pump is not affected by the system, it is easy to determine if a positive displacement pump has worn internals. This fact is shown in Figure 2.10.

Positive displacement pump internal wear

Is identified by reduced flow rate

- Control valve closing
- Reduced amps

Figure 2.10 Positive displacement pump internal wear

Centrifugal (Kinetic) pumps

Centrifugal pumps increase the pressure of the liquid by using rotating blades to increase the velocity of a liquid and then reduce the velocity of the liquid in the volute. Refer again to Figure 2.5.

A good analogy to this procedure is a football (soccer) game. When the ball (liquid molecule) is kicked, the leg (vane) increases its velocity. When the goal tender (volute), hopefully, catches the ball, its velocity is significantly reduced and the pressure in the ball (molecule) is increased. If an instant replay 'freeze shot' picture is taken of the ball at this instant, the volume of the ball is reduced and the pressure is increased.

The characteristics of any centrifugal pump then are significantly different from positive displacement pumps and are noted in Figure 2.11.

Centrifugal pump characteristics

- Variable flow
- Fixed pressure produced *for a specific flow*
- Does not require a pressure limiting device
- Flow varies with differential pressure (P_2-P_1) and/or specific gravity

Figure 2.11 Centrifugal pump characteristics

Refer again to Figure 2.8 and note that all pumps react to the process requirements.

Based on the characteristics of centrifugal pumps noted in Figure 2.11, the flow rate of all types of centrifugal pumps is affected by the process system. This fact is shown in Figure 2.12.

Therefore, the flow rate of any centrifugal pump is affected by the system.

Refer to Figure 2.13 and it can be observed that all types of mechanical failures can occur based on *where the pump is operating based on the process requirements.*

Since greater than 95% of the pumps used in any plant are centrifugal, their operating flow will be affected by the process.

Important facts concerning this failure classification are noted in Figure 2.14.

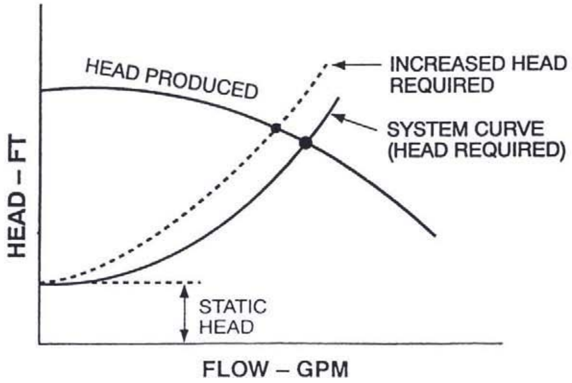


Figure 2.12 A centrifugal pump in a process system

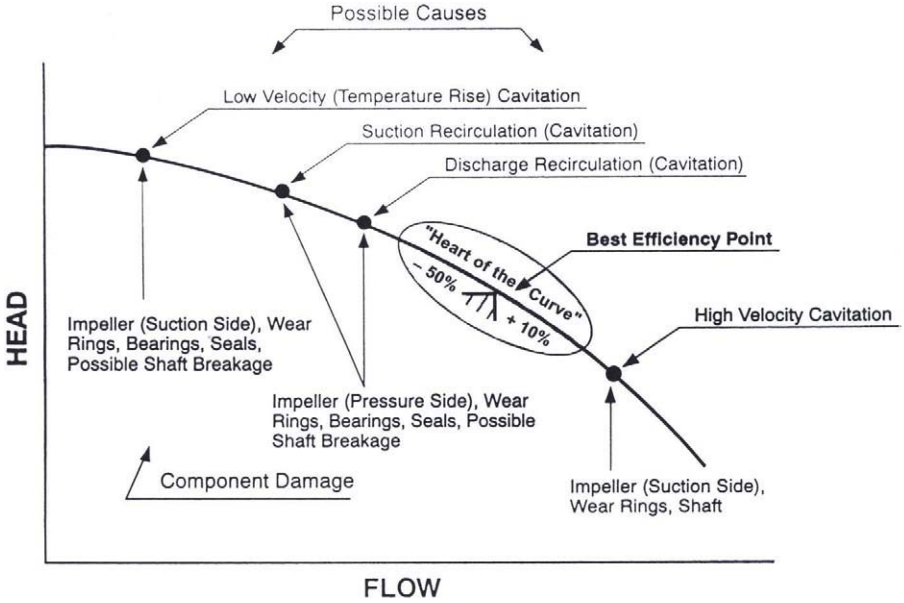

Possible Causes
Low Velocity (Temperature Rise) Cavitation
Suction Recirculation (Cavitation)
Discharge Recirculation (Cavitation)
"Heart of the Curve"
– 50%
+ 10%
Best Efficiency Point
High Velocity Cavitation
HEAD
Impeller (Suction Side), Wear Rings, Bearings, Seals, Possible Shaft Breakage
Impeller (Pressure Side), Wear Rings, Bearings, Seals, Possible Shaft Breakage
Component Damage
Impeller (Suction Side), Wear Rings, Shaft
FLOW

Figure 2.13 Centrifugal pump component damage and causes as a function of operating point

Centrifugal pump reliability

- Is affected by process system changes (system resistance and S.G.)
- It is not affected by the operators
- Increased differential pressure (P_2–P_1) means reduced flow rate
- Decreased differential pressure (P_2–P_1) means increased flow rate

Figure 2.14 Centrifugal pump reliability

At this point it should be easy to see how we can condition monitor the centrifugal pump operating point. Refer to Figure 2.15.

Centrifugal pump practical condition monitoring

- Monitor flow and check where the operating point is on the shop test curve.
- Flow can also be monitored by:
  - Control valve position
  - Motor amps
  - Steam turbine valve position
  - Differential pipe temperature (outlet-inlet)

Figure 2.15 Centrifugal pump practical condition monitoring

The definitions and characteristics of positive displacement and dynamic equipment are presented in Figure 2.16.

Positive displacement – Dynamic pump comparison

	Positive displacement	Dynamic
Definition	Increase pressure by operating on a fixed volume in a confined space	Increases pressure by using rotary blades to increase fluid velocity
Types	Screw, gear, reciprocating	Centrifugal axial
Characteristics	■ Constant volume ■ Variable differential head ■ Relatively insensitive to liquid properties ■ Relatively insensitive to system changes ■ Not self-limiting	■ Variable volume ■ Constant differential head ■ Sensitive to liquid properties ■ Sensitive to system changes ■ Self-limiting

Characteristic flow vs. differential head curves

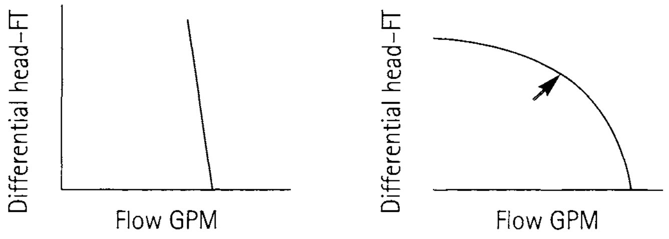


Figure 2.16 Positive displacement – dynamic pump comparison

Driver reliability (motors, steam turbine and diesel engines) can also be affected by the process when centrifugal driven equipment (pumps, compressor and fans) are used.

Refer to Figure 2.17 and observe typical centrifugal pump curve.

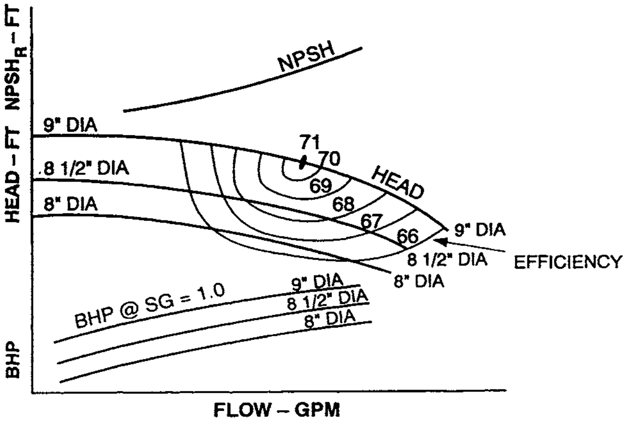


Figure 2.17 A typical centrifugal pump performance curve

Since the flow rate will be determined by the process requirements, the power (BHP) required by the driver will also be affected. What would occur if an $8^1/2$" diameter impeller were used and the head (differential pressure) required by the process was low? Answer: Since the pressure differential required is low, the flow rate will increase and for the $8^1/2$" diameter impeller, the power required by the driver (BHP) will increase.

Therefore, a motor can trip out on overload, a steam turbine's speed can reduce or a diesel engine can trip on high engine temperature. These facts are shown in Figure 2.18.

Effect of the process on drivers

- Motors can trip on overload
- Steam turbines can reduce speed
- Diesel engines can trip on high engine temperature

Figure 2.18 Effect of the process on drivers

Auxiliary system reliability is also affected by process changes. Auxiliary systems support the equipment and their components by providing ... clean, cool fluid to the components at the correct differential pressure, temperature and flow rate.

Typical auxiliary systems are:

- Lube oil system
- Seal flush system
- Seal steam quench by system
- Cooling water system

The reliability of machinery components (bearings, seals, etc.) is directly related to the reliability of the auxiliary system. In many cases, the root cause of the component failure is found in the supporting auxiliary system.

As an example, changes in auxiliary system supply temperature, resulting from cooling water temperature or ambient air temperature changes, can be the root cause of component failure. Figure 2.19 presents these facts.

Component (bearing and seal) reliability

- Is directly related to auxiliary system reliability.
- Auxiliary system reliability is affected by process condition changes.
- 'Root causes' of component failure are often found in the auxiliary system.

Figure 2.19 Component (bearing and seal) reliability

As a result, the condition of all the auxiliary systems supporting a piece of equipment must be monitored. Please refer to Figure 2.20.

Always 'Think system'

- Monitor auxiliary system condition
- Inspect auxiliary systems during component replacement

Figure 2.20 Always 'Think system'

2. Improper assembly maintenance/installation

In proper assembly, maintenance (lubrication) of components and/or improper installation practices will shorten the life of components and cause eventual failure because the anticipated design factors were not met.

Tolerances, maintenance requirements and installation procedures are provided to assure maximum component and equipment life. As an example, refer to Figure 2.21, which shows anti-friction bearings commonly used in pumps.

The relationship that determines how long an anti-friction bearing will last is shown in Figure 2.22.

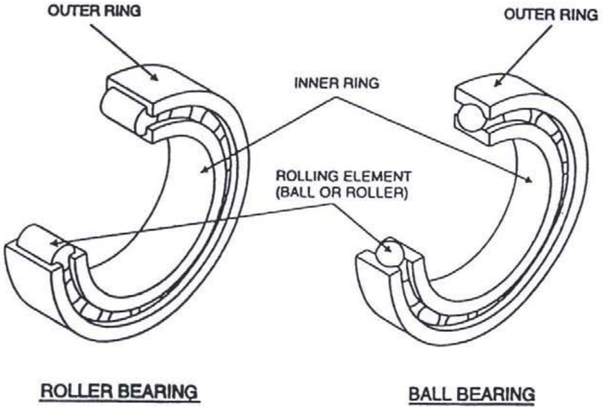

OUTER RING
OUTER RING
INNER RING
ROLLING ELEMENT (BALL OR ROLLER)
ROLLER BEARING
BALL BEARING

Figure 2.21 Anti-friction bearing

L-10 LIFE

'B' or 'L' - 10 Life is defined as the life in hours that 9 out of 10 randomly selected bearings would exceed in a specific application.

$$\text{'B' or 'L' - 10 Life} = \frac{16700}{N}\left[\frac{C}{F}\right]^3$$

Where: N = RPM
C = Load in LBS that will result in a bearing element life of 1,000,000 revolutions.
F = Actual load in LBS

Figure 2.22 L-10 LIFE

Note that the life of the bearing is directly dependent on the forces acting on the bearing to the 3rd power. As an example, if the forces were twice the design value, the life of the bearing would be reduced eight times! The minimum specified life for a bearing is three years or 36 months. In this example, the life would be reduced to approximately five months!

What can cause excessive bearing forces in this case? Refer to Figure 2.23.

Sources of forces

- Increased process pipe forces and moments
- Foundation forces ('soft' foot, differential settlement)
- Fouling of plugging of impeller
- Misalignment
- Unbalance
- Rubs
- Improper assembly clearances
- Thermal expansion of components (loss of cooling medium, excessive operating temperature)
- Radial forces (single volute – off design operation)
- Poor piping layouts (causing unequal flow distribution to the pump)

Figure 2.23 Sources of forces

When any bearing is designed, it has a maximum acceptable total force which will allow it to operate trouble-free for a period equal to, or in excess of, its specified life. If the total forces acting on the bearing exceed this value, there will be a bearing failure.

Therefore, the bearing must be installed in accordance with the vendor's instructions as detailed in the instruction book. The use of general procedures or 'rules of thumb' (using typical values) should be avoided.

In addition to the component assembly procedure, the pump installation procedure must be followed. The pump installation procedure assures that the following items are checked as noted in Figure 2.24.

Pump installation procedure requirements

- Proper pump/driver alignment
- Minimum external pipe forces
- Minimum external foundation forces ('soft foot')

Figure 2.24 Pump installation procedure requirements

Now refer back to Figure 2.23 and observe what additional forces, not anticipated by the vendor, that can be added to the total bearing force!

Therefore, improper installation procedure values will cause bearing failure.

In summary, the proper steps to prevent assembly and/or installation errors are presented in Figure 2.25.

Avoid assembly/installation errors by ...

- Having the instruction book available
- Completely following specified procedures
- Using only specified parts
- Using refinery procedures for:
  - Alignment
  - Pipe stress check
  - Soft foot check

Figure 2.25 Avoid assembly/installation errors by ...

A final word of advice regarding procedures is given in Figure 2.26.

Remember!!

Exceed the specified limits ... and you will reduced equipment reliability

Figure 2.26 Procedures – advice

The final step in any assembly and/or installation procedure is to confirm that the procedure was performed properly. This is accomplished by condition monitoring of the replaced components after the machine is operating at normal conditions ... 'lined out'. This method is outlined in Figure 2.27.

Post component replacement check guidelines

- Obtain component condition data
- Compare data to:
  - Data before component replacement
  - Site guidelines
  - Confirm all component condition data is satisfactory (sign-off)

Figure 2.27 Post component replacement check guidelines

A good question at this point would be ... How long after 'line out' should this check be performed? The norm is during the first 4–8 hours after normal operation 'line out' is attained.

All rotating equipment, when manufactured, is only tested for approximately 4 hours. In the case of a pump operating at 3600 RPM, this pump will have rotated 1,000,000 times during this period! If the assembly/installation procedure requirements were not met, we will know from the guidelines at this time.

3. Improper operating procedures

Failure of machinery and/or components can occur because equipment will be subjected to conditions that exceed the design values. Refer to Figure 2.28.

Improper operating procedures

- Subject the equipment to conditions that exceed design value limits.

Figure 2.28 Improper operating procedures

Operating procedures can be the root cause of failure. Please refer to Figure 2.29.

Operating procedures can cause failure if

- They are not complete
- They are not followed
- The actual operating conditions are different than specified

Figure 2.29 Operating procedures can cause failure if ...

Most machinery and/or component damage and wear occur during start-up or shutdown (transient) conditions. During this time, the equipment is subjected to rapid temperature, pressure and speed changes.

Shown in Figure 2.30 are some examples of operating procedure requirements, the reason for the requirement, the consequences, the checks and corrective action.

Operating procedure requirements

Requirement	Reasons	Affected components	Check(s) to prevent problem	Corrective action
1. Pump & seal vented and full of liquid	Assure only liquid is present to remove frictional heat from close running parts, assure continuous pumping and prevent immediate seal failure	Casing, impeller(s), shaft, wear rings, seal(s) and couplings	That discharge pressure is reached immediately and does not fluctuate	Check process system to determine cause. If discharge pressure does not build up, shut down pump and investigate
2. Pump suction valve wide open, no suction line restrictions	Same as item 1, prevent cavitation and assure discharge pressure is reached	Same as item 1	Suction valve wide open, discharge pressure reached and does not fluctuate, no cavitation noise	Confirm suction valve is wide open. Shutdown pump if problem remains check suction strainer (suction basket if sump pump). Look for suction line obstructions

Requirement	Reasons	Affected components	Check(s) to prevent problem	Corrective action
3. Discharge pump valve pinched for start-up[1]	Prevent pump from running dry. Prevent high flow cavitation. Prevent driver overload and high inrush motor current (which will reduce insulation life)	Same as item 1	Discharge valve is not full open. Discharge pressure is reached quickly, with a steady rise in pressure. No cavitation noise	Confirm discharge valve is not full open. Shutdown pump if problem remains. Re-start with discharge valve partially closed. Fully open discharge after pump has reached full speed
4. Steam turbine & inlet steam line warmed.	Prevent slugging the turbine with condensate.	Bearings, steam seals and possibly turbine blades	Check drains for presence of condensate and check that steam is above saturation temperature	Open drain lines until condensate flow stops. Start turbine slowly, using small bypass valve supplied and confirm absence of condensate.
5. Standby hot service pump warmed.	Reduces thermal shock, assures proper liquid viscosity and correct shaft alignment.	Pump casing, seals, bearings and coupling.	Check temperature of casing, seal area and suction line	Confirm bypass valve around the discharge check valve and suction valve are open.
6. Cold service pump chilldown.	Eliminates vapor in pump case and seal.	Wear rings and seals.	That discharge pressure is reached immediately. Vent casing high point and seal chamber.	Shut down pump if discharge pressure is not reached. Open vent lines of casing and seal chamber and confirm fluid in unit is 100% liquid.
7.Steam turbine slow roll & start-up sequence.	Reduces thermal shock & assures correct alignment.	Casing, rotor, internal seals, shaft end seals, coupling and bearings.	Check drains for presence of condensate and check that steam is above saturation temperature	Prior to slowroll, drain casing, throttle valve and inlet line. Do not commission steam seal system (if supplied) until turbine is turning. Completely follow vendor's start-up instructions.

[1] For process unit start-up when no other pumps are in operation

Figure 2.30 Operating procedure requirements

How can failures associated with operating procedures and their implementation be avoided?

Figure 2.31 presents some guidelines.

Operating procedure reliability guidelines

- Confirm the operating conditions are as specified
- Confirm the refinery instruction manual 'RIM' is in agreement with vendor's instructions
- Understand the reason for the requirement
- Do not hesitate to ask ...

Figure 2.31 Operating procedure reliability guidelines

The importance of having operating procedures that are accurate, properly written and completely followed cannot be overemphasized. As previously stated, the transient (rapid change) conditions that equipment is exposed to during start-up and shutdown can cause rapid component wear or failure. These facts are presented in Figure 2.32.

Follow procedures completely because ...

- During start-up and shutdown, equipment components are subject to rapid (transient)
  - Temperature changes
  - Pressure changes
  - Speed changes.
- Remember, most component wear occurs during transient conditions.

Figure 2.32 Follow procedures completely because ...

4. Design problems

Possible design problems manifest themselves like all other causes of failure ... component condition values are exceeded. Before a design problem is confirmed, three previously discussed causes of failure should be checked as shown in Figure 2.33.

Possible design problem?

- First confirm the following classifications are not the root cause:
  - Process condition change
  - Assembly/installation procedures
  - Operating procedures

Figure 2.33 Possible design problem?

Design problems can fall into three categories as shown in Figure 2.34.

Design problem categories

- Engineering errors
- Material problems
- Manufacturing problems

Figure 2.34 Design problem categories

Design problems usually show up shortly after the process unit is at normal conditions – 'line out'. However, there are cases when design problems manifest themselves after extended operation and even the warranty period.

The cause of design problems is that the machine and/or its components were not designed for the *specified* field operating conditions. See Figure 2.35.

A design problem exists if:

- The machine and/or components are not designed for *specified* field operating conditions.

Figure 2.35 A design problem exists if:

Often, the vendor is accused of a design error when, in fact, the specified conditions and/or operating procedures have changed.

After 'line out' of the process, if problems exist (component condition

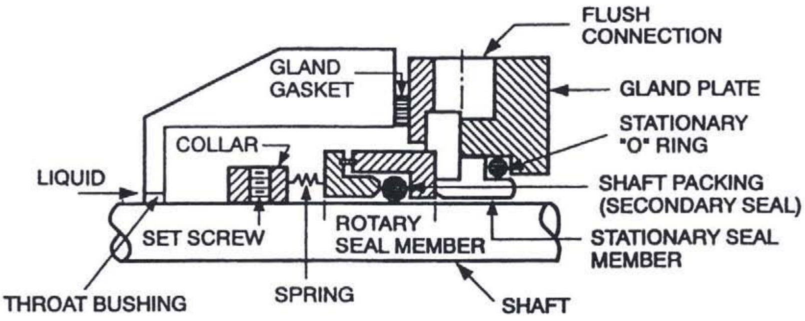

FLUSH CONNECTION
GLAND GASKET
GLAND PLATE
STATIONARY "O" RING
COLLAR
LIQUID
SHAFT PACKING (SECONDARY SEAL)
SET SCREW
ROTARY SEAL MEMBER
STATIONARY SEAL MEMBER
THROAT BUSHING
SPRING
SHAFT

Figure 2.36 Pump mechanical seal

values exceeded), the equipment data sheet should be compared to the actual data to confirm the equipment was designed to the actual field operating conditions. If not, field conditions should be corrected if possible. If field conditions remain different than specified on the data sheet, it is not a design problem, it is an 'application problem'. In this situation, the vendor is justified in asking for redesign costs if necessary.

An example of a possible design problem is a leaking mechanical seal. Refer to Figure 2.36.

All mechanical pump seals are designed to convert the liquid to a vapor across the seal face. If the actual operating liquid conditions (vapor pressure, temperature and pump pressures) are not specified, a mechanical seal failure can occur because either the liquid is not changed to a vapor or the liquid vaporizes in the seal chamber (stuffing box).

Figure 2.37 shows the change of pressure and temperature across a seal face in three (3) cases:

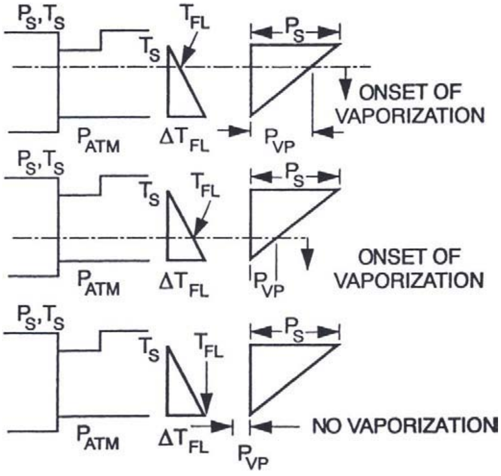

P_S, T_S
T_{FL}
T_S
P_S
ONSET OF VAPORIZATION
P_{ATM}
ΔT_{FL}
P_{VP}
P_S, T_S
T_S
T_{FL}
P_S
ONSET OF VAPORIZATION
P_{ATM}
ΔT_{FL}
P_{VP}
P_S, T_S
T_S
T_{FL}
P_S
P_{ATM}
ΔT_{FL}
NO VAPORIZATION
P_{VP}

Figure 2.37 Seal face vaporization

- Early vaporization
- Proper design
- No vaporization

In this example, a pump 'bad actor', with more than one (1) seal failure per year, should first be checked for proper liquid conditions at the seal face before it is classified as a mechanical seal design problem. The cause of failure may be a process related issue (improper liquid conditions or plugged flush line orifice). If all conditions are as specified, then it is truly a mechanical design problem.

Another example of a possible design problem is oil contamination in the bearing brackets of a single stage steam turbine shown in Figure 2.38. Assume continuous problems are experienced with water in the oil causing bearing failures. It is also confirmed that the source of water is from the carbon ring seal leakage of steam into the bearing bracket.

As was previously stated, first check the other failure causes:

- Process condition changes
- Assembly/installation procedures
- Operating procedures

A check confirms that the failure causes noted above did not occur.

Therefore, the carbon ring seal system (carbon ring seals and bearing bracket isolator) are not designed to prevent oil contamination in the bearing bracket. This case is an example of a true design problem. For this example, there are two (2) possible modifications:

- Install an eductor system to positively prevent steam leakage from the seal assembly (presently a requirement of a major oil company general purpose steam turbine spec.).
- Install a bearing isolator to positively prevent steam condensate from entering the bearing bracket ('Impro' type of equal).

In summary, the factors concerning possible design problems are noted in Figure 2.39.

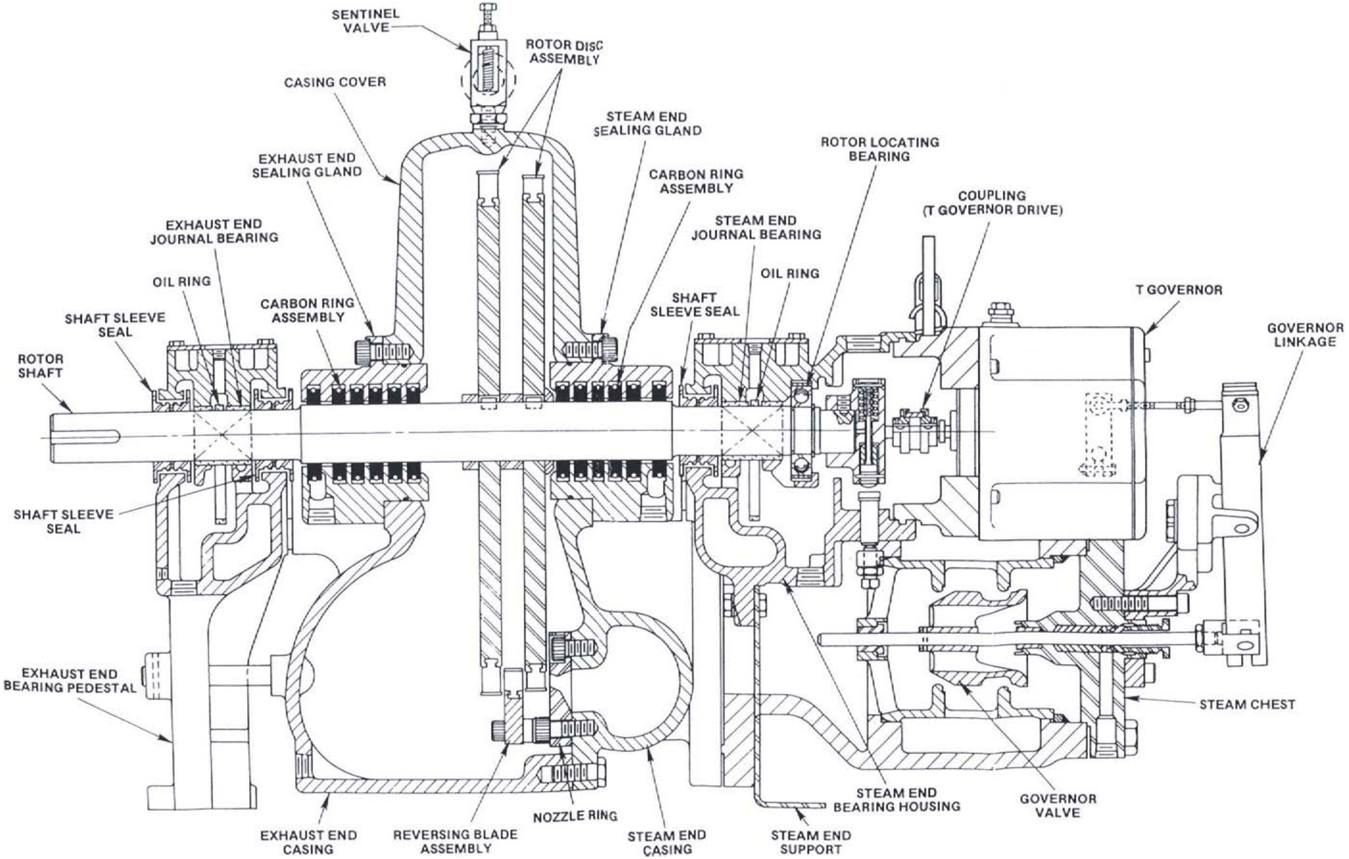

SENTINEL VALVE
ROTOR DISC ASSEMBLY
CASING COVER
STEAM END SEALING GLAND
ROTOR LOCATING BEARING
EXHAUST END SEALING GLAND
CARBON RING ASSEMBLY
COUPLING (T GOVERNOR DRIVE)
STEAM END JOURNAL BEARING
EXHAUST END JOURNAL BEARING
OIL RING
OIL RING
T GOVERNOR
SHAFT SLEEVE SEAL
CARBON RING ASSEMBLY
SHAFT SLEEVE SEAL
GOVERNOR LINKAGE
ROTOR SHAFT
SHAFT SLEEVE SEAL
EXHAUST END BEARING PEDESTAL
STEAM CHEST
STEAM END BEARING HOUSING
GOVERNOR VALVE
NOZZLE RING
EXHAUST END CASING
REVERSING BLADE ASSEMBLY
STEAM END CASING
STEAM END SUPPORT

Figure 2.38 Typical general purpose (pump drive) turbine assembly (Courtesy of the Elliott Company)

Design problem determination and action plan

- Confirm all other failure causes do not exist
- Confirm specified operating conditions exist in the field (check data sheet)
- Conduct a design audit meeting with the vendor (if necessary)

Figure 2.39 Design problem determination and action plan

5. Component wearout

Like design problems, component wearout is often determined to be a root cause of failure.

However, as shown in Figure 2.40, apparent component wearout usually is caused by other failure classifications.

Apparent component wearout

- Usually is caused by one or more of the following failure classifications:
  - Process condition changes affecting the equipment and/or its auxiliary systems
  - Assembly/maintenance/installation errors
  - Improper operating procedures
  - Design deficiencies

Figure 2.40 Apparent component wearout

Always investigate the other failure classifications first. In many cases, component wearout is the effect, not the cause of the problem. Figure 2.41 presents this information.

Component wearout is often the effect, not the root cause of the problem

Figure 2.41

Refer back to failure classification 1, process condition changes Figure 2.13. What components could 'wearout' if the process required high differential pressure?

Also refer to failure classification 2, assembly, installation problems, Figures 2.21, 2.22 and 2.23. Why could a bearing ‘wearout’ if the foundation cracked? Why would the bearing ‘wearout’ if a shim fell out of pipe support?

Finally, refer to failure classification 3, improper operating procedures, Figure 2.30. Why could a seal with a flush from its discharge line wearout if a loading pump were started with the discharge valve wide open?

These examples are presented in Figure 2.42.

Apparent component wearout examples

Component	Root Cause
Bearing, seal, impeller wear ring	Process condition change
Bearing	Installation problems
Mechanical seal	Operating procedure (pump run dry)

Figure 2.42 Apparent component wearout examples

Therefore, when will equipment components wear out? If all the considerations to eliminate failure classifications are met, components will last a long time.

Think of some site pumps and name the shortest and longest periods between:

- Bearing replacement
- Seal replacement

Figure 2.43 presents some components and their life if failure classifications 1–4 are not present.

Component Life

(failure classifications 1–4 not present)

Component	Life
Anti-friction bearing	8–10 years
Sleeve bearings	15–20 years
Mechanical seal	7–10 years
Wear rings	10–12 years
Impellers	15–20 years

Figure 2.43 Component life

In many cases, component wearout is a result of the wearout of the 'secondary' parts in the component. An example is 'o' rings in mechanical seals.

As in the case of the previous failure classifications, component wearout does not randomly occur. The condition parameters associated with these components will change. Figure 2.44 presents the guidelines to determine component wearout.

Component wearout guidelines

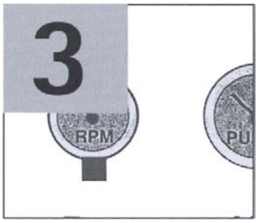
- Monitor component condition parameters
- Plan scheduled shutdown using predictive maintenance (PDM) principles
- Confirm failure classifications 1–4 are not present.

Figure 2.44 Component wearout guidelines

Summary

At this point the five (5) failure classifications have been discussed in detail. How can these failures be minimized?

The next chapter, 'How to prevent machinery failures', will discuss this subject. The answers should be clear at this point.



3

How to prevent machinery failures

- Introduction
- Component function awareness – ‘What should it do?’
- Component condition monitoring – ‘What is it doing?’
- Preventive (PM) and predictive maintenance (PDM)
- Troubleshooting
- Reliability, everyone’s responsibility

Introduction

The five (5) machinery failure classifications are presented in Figure 3.1.

Failure classifications

- Process condition changes
- Improper assembly/maintenance/installation
- Improper operating procedures
- Design deficiencies
- Component wearout

Figure 3.1 Failure classifications

The details concerning each of these failure classifications were discussed in the previous chapter. How can these failure causes be prevented?

Re-examination of the details concerning each failure classification shows that the solution to the prevention of each failure cause is identical. This fact is presented in Figure 3.2.

Prevent machinery failures by ...

- Component function awareness (what should it do?)
- Component condition monitoring (what is it doing?)
- Using predictive maintenance techniques
- Teamwork – reliability is everyone's responsibility

Figure 3.2 Prevent machinery failures by ...

Let's now examine each of the action items noted above in detail.

Component function awareness – 'What should it do?'

Component (machinery part) function awareness allows you to determine what the component is supposed to do. It is obvious that a certain amount of knowledge is required to accomplish this fact. Remember, you may not have all of the knowledge required. **OBTAIN IT!** Figure 3.3 presents sources of where the information may be obtained.

What is it supposed to do?

- Thoroughly understand the function of each component by:
- Reading the instruction book
- Asking questions of:
  - Site reliability group
  - Site technical group
  - Machinists
  - Operators
- Referring to reference books
- Organizing 'mini' information sessions for operators and machinists

Figure 3.3 What is it supposed to do?

Figure 3.4 presents the important principle of knowledge base. The greater this base, the more effective predictive maintenance and root cause analysis procedures will be.

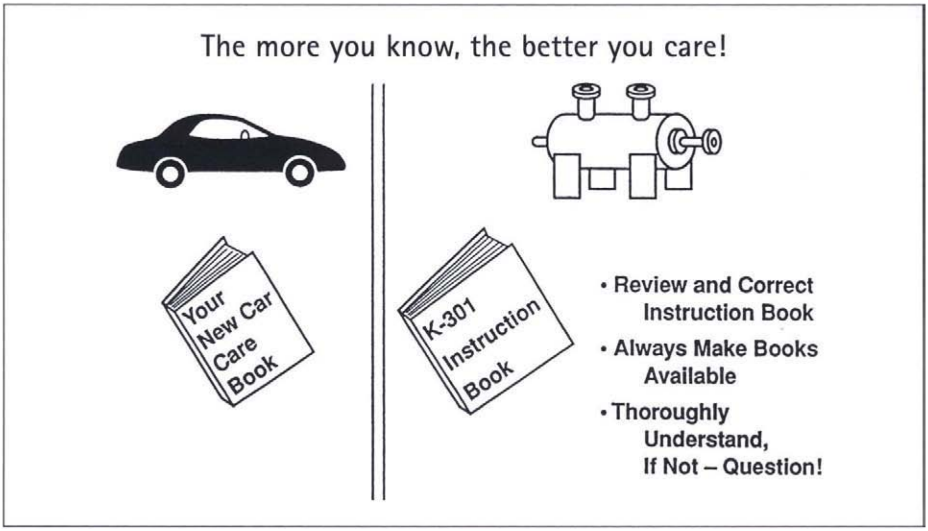

The more you know, the better you care!
Your New Car Care Book
K-301 Instruction Book
• Review and Correct Instruction Book
• Always Make Books Available
• Thoroughly Understand, If Not – Question!

Figure 3.4 The more you know, the better you care!

One final word. Do not be afraid to admit to management that you do not know certain aspects of a problem. But be sure to state that you will find out. After all, management must understand that this is a learning process and does require time.

To aid in the understanding of component function definition, we have included an example for an anti-friction bearing in Figure 3.5.

Component function example

An anti-friction bearing continuously supports all *static and dynamic forces* of a rotor by providing sufficient *bearing area* and requires *oil flow* to remove the generated *frictional heat.*

This statement then defines the items that must be monitored to determine the bearing's condition:

- Static and dynamic forces
- Bearing area
- Oil flow
- Frictional heat

Figure 3.5 Component function example

Naturally, we cannot measure directly all of the items noted in Figure 3.5. However, based on the instruments and measuring devices available on site, what can be measured to assure the component (bearing) is performing correctly?

Component condition monitoring – 'What is it doing?'

In reference to the anti-friction bearing example, the component condition monitoring parameters are presented in Figure 3.6.

Component condition monitoring parameters (for anti-friction bearing)

- Bearing housing vibration
- Bearing housing temperature
- Lube oil condition
  - Viscosity
  - Water content
  - Oil particle content

Figure 3.6 Component condition monitoring parameters (for anti-friction bearing)

A similar exercise can be conducted for all of the major components and systems in any piece of equipment.

What are the major components and systems of any piece of rotating equipment? How many are there? And are the same components contained in any type of rotating equipment? The answers to these questions will be discussed in a later chapter of this book and form the principle of component condition monitoring.

In the next chapter, all of the parameters that should be monitored for each machinery component and its systems regardless of the type will be defined.

Preventive (PM) and predictive maintenance (PDM)

At this point, the distinctions between preventive maintenance (PM), predictive maintenance (PDM) and troubleshooting must be discussed.

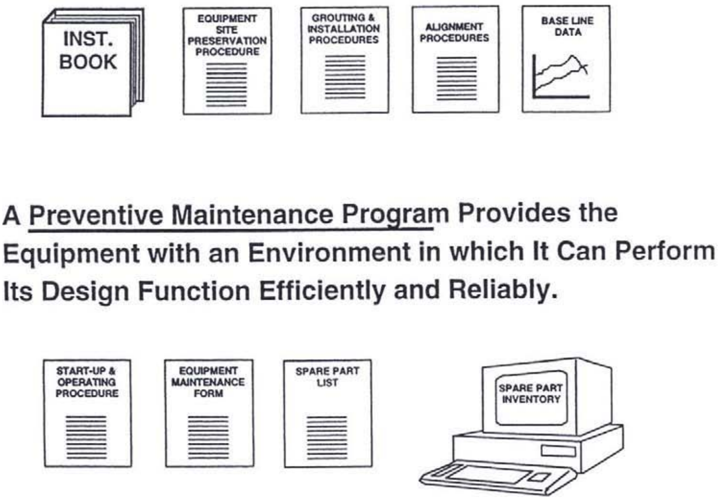

INST. BOOK
EQUIPMENT SITE PRESERVATION PROCEDURE
GROUTING & INSTALLATION PROCEDURES
ALIGNMENT PROCEDURES
BASE LINE DATA
A Preventive Maintenance Program Provides the Equipment with an Environment in which It Can Perform Its Design Function Efficiently and Reliably.
START-UP & OPERATING PROCEDURE
EQUIPMENT MAINTENANCE FORM
SPARE PART LIST
SPARE PART INVENTORY

Figure 3.7 A typical preventive maintenance program (Courtesy of M.E. Crane Consultant)

Preventive maintenance

Preventive maintenance requires that maintenance be performed at predetermined intervals. It is time based. A most common preventive maintenance step is an automotive oil change. The objective of this action is to remove the oil from the engine before oil contamination and deterioration cause excessive wear to the engine components. Figure 3.7 presents the components of a typical site preventive maintenance program.

In our experience, a well-planned preventive maintenance program can truly be effective. However, the question must be asked, ‘Is the maintenance performed always necessary?’ Refer to Figure 3.8.

Preventive maintenance

- Preventive maintenance prevents but ... takes time.
- Is it always necessary?

Figure 3.8 Preventive maintenance

What is the basis for replacing components? Unnecessary component replacement exposes the machinery unit to a failure classification (improper assembly of components, improper installation, component malfunction, component improper storage procedures, etc.).

In addition, preventive maintenance can cause a mindset that automatically determines maintenance at every turnaround regardless of component condition. This can be a costly practice. A case history also demonstrates where preventive maintenance can lead if not properly monitored.

A centrifugal compressor in a large refinery was scheduled for maintenance during the upcoming turnaround. Maintenance planning had scheduled bearing inspection and change if necessary. During the turnaround when bearings were inspected, excessive clearances and signs of deterioration were found. Naturally the bearings were replaced. However, because the bearings were replaced, it was decided that the seals, which are more difficult to remove and inspect, be observed. Upon seal removal the seals were also in a distressed condition and needed to be replaced. Now the tough decisions had to be made. It was decided that the compressor would be dissembled to inspect interior condition for possible causes of seal and bearing failure. Upon disassembly, no significant abnormalities were found within the compressor and it was consequently reassembled.

This case history demonstrates how a standard preventive maintenance approach can lead to unnecessary maintenance and significant loss of revenue to the operating unit. In this case, the operating unit did not make use of site instrumentation. Nowhere had people answered the question 'What changed?'. This approach therefore led to unnecessary disassembly of the compressor. If bearing parameters (temperature, vibration, etc.) and seal parameters (inner and outer seal leakage) had been monitored for change, the conclusions of only bearing and seal change would have been made without unnecessary disassembly. Remember, to disassemble a compressor, significant additional tools and materials are required. Typical time for compressor disassembly can easily reach one week. It can be seen therefore, that the effective way to perform any maintenance activity is to thoroughly plan that activity based on condition changes to equipment. This leads us to the discussion of predictive maintenance.

Predictive maintenance

Predictive maintenance is based on component condition monitoring and trending. Figure 3.9 presents the definition of predictive maintenance.

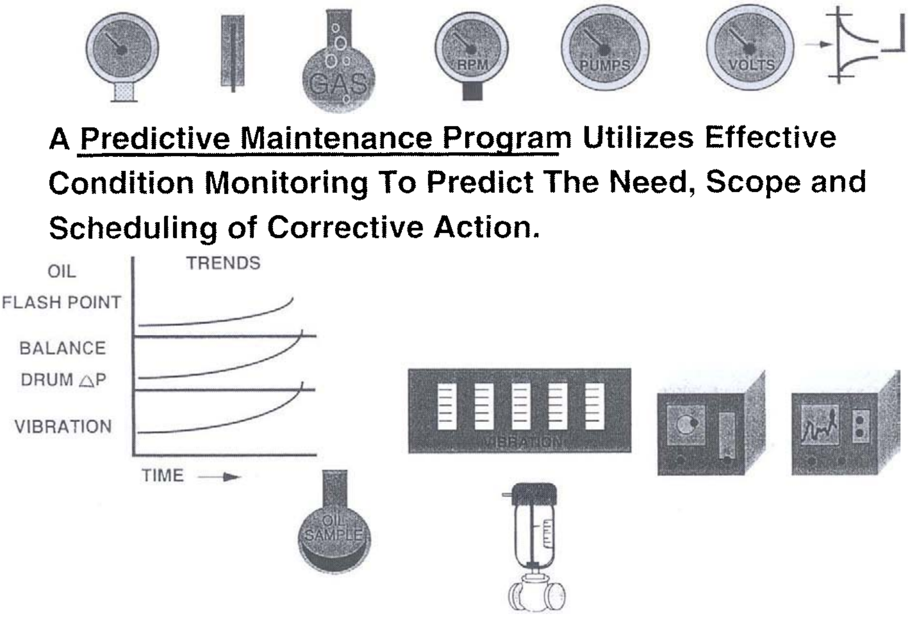

GAS
RPM
PUMPS
VOLTS
A Predictive Maintenance Program Utilizes Effective Condition Monitoring To Predict The Need, Scope and Scheduling of Corrective Action.
OIL
FLASH POINT
TRENDS
BALANCE
DRUM ΔP
VIBRATION
TIME
VIBRATION
OIL SAMPLE

Figure 3.9 A predictive maintenance program (Courtesy of M.E. Crane Consultant)

Troubleshooting

Wherever I travel, worldwide, Troubleshooting is the 'keyword'. More recently, other 'keywords' have emerged:

- Failure analysis
- Root cause analysis
- Reliability centered maintenance (RCM)

Regardless of the 'keyword', it's still troubleshooting. This term is defined in Figure 3.10.

Definition

Troubleshoot – to **discover** and **eliminate (root)** causes of trouble

Figure 3.10 Definition

What are the requirements to accomplish an effective troubleshooting exercise? These facts are presented in Figure 3.11.

- Troubleshooting requires that all **abnormal** conditions be defined
- However, to determine **abnormal** conditions, the **normal** conditions must be known
- Therefore **baseline (normal)** conditions must be known

Figure 3.11 An effective troubleshooting exercise

Do these requirements sound familiar? They certainly should. These are the requirements for predictive maintenance! The differences between these two terms are presented in Figure 3.12.

Predictive maintenance and troubleshooting

- **Predictive maintenance** requires baseline and trend data to predict the root cause of the **change in condition**.
- **Troubleshooting** requires baseline and trend data to predict the root cause of **failure**

Figure 3.12 Predictive maintenance and troubleshooting

Therefore, if we use site-wide predictive maintenance techniques, we can *potentially* detect a *change in condition* before failure. Please refer to Figure 3.13.

Troubleshooting ...

Is predictive maintenance after a failure!

Figure 3.13 Troubleshooting ...

Notice that in the discussion above, the word 'potentially' was in italics.

Remember that the majority of rotating equipment in any plant is general purpose or spared equipment that is not continuously monitored in the control room DCS system. This equipment is also the source of most reliability problems ('Bad Actors'). How can this equipment be effectively monitored?

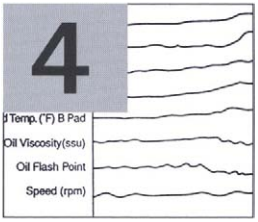
Let's now discuss the final topic in this chapter.

Reliability, everyone's responsibility

Please refer again to Figure 3.4 of this chapter and review the analogy between your vehicle and site machinery. You'll have fewer problems if you and the mechanic (operators and machinists) know more and – work as a team!

Reliability must be everyone's responsibility. The entire plant operations, maintenance and engineering departments must be aware of the reliability program philosophy and must be able to implement it. Having operators and machinists equipped and trained in the use of simple vibration instruments (vibration pens), oil condition monitors and laser temperature guns will significantly increase the reliability of general purpose (spared) equipment thru the implementation of an effective predictive maintenance program.

4



Optimizing CCM and PDM

(Component condition monitoring and predictive maintenance)

- The major machinery components
- Component condition monitoring
- Predictive maintenance (PDM) techniques

The major machinery components

Think of all the machinery that you have been associated with and ask ... What are the major components and systems that are common to all types of rotating equipment?

Figure 4.1 presents the major component classifications for any type of machinery:

- Pumps
- Steam turbines
- Compressors
- Motors
- Gas turbines
- Fans
- Etc.

Major machinery components and systems

- Rotor
- Radial bearing
- Thrust bearing
- Seal
- Auxiliary systems

Figure 4.1 Major machinery components and systems

Regardless of the type of machinery, monitor these components and you will know the total condition of the machine.

Component condition monitoring

As previously stated, component and system functions must first be defined and the normal values for each component listed. These facts are presented in Figure 4.2.

Component and system functions

- Define the function of each affected component
- Define the system in which each affected component operates
- List the normal parameters for each affected component and system component

Figure 4.2 Component and system functions

Once the function of each component is defined, each major machinery component can be monitored as shown in Figure 4.3.

Component condition monitoring

- **Define** each major component
- **List** condition monitoring parameters
- **Obtain** baseline data
- **Trend** data
- **Establish** threshold limits

Figure 4.3 Component condition monitoring

Baseline

Having defined all condition parameters that must be monitored, the next step in a condition monitoring exercise is to obtain baseline information. It is important to obtain baseline information as soon as physically possible after start-up of equipment. However, operations should be consulted to confirm when the unit is operating at rated or lined out conditions. Obtaining baseline information without conferring with operations is not suggested since mis-information could be obtained and thus lead to erroneous conclusions in predictive maintenance. Figure 4.4 states the basics of a baseline condition.

Base line condition

If you don't know where you started, you do not know where you are going!

Figure 4.4 Base line condition

It is amazing to us how many times baseline conditions are ignored. Please remember Figure 4.4 and make it a practice to obtain baseline conditions as soon as possible after start-up.

Trending

Trending is simply the practice of monitoring parameter condition with time. Trending begins with baseline condition and will continue until equipment shutdown. In modern day thought, it is often conjectured that trending must be performed by micro-processors and sophisticated control systems. This is not necessary! Effective trending can be obtained by periodic manual observation of equipment or using equipment available to us in the plant which will include DCS systems, etc. The important fact is to obtain the baseline and trends of data on a periodic basis. When trending data, threshold points should also be defined for each parameter that is trended. This means that when the parameter pre-established value is exceeded action must be taken regarding problem analysis. Setting threshold values a standard percentage above normal value is recommended. Typically values are on the order of 25–30% above baseline values. However, these values must be defined for each component based on experience. Figure 4.5 presents trending data for a hydrodynamic journal bearing. All of the parameters noted in Figure 4.5 should be monitored to define the condition of this journal bearing.

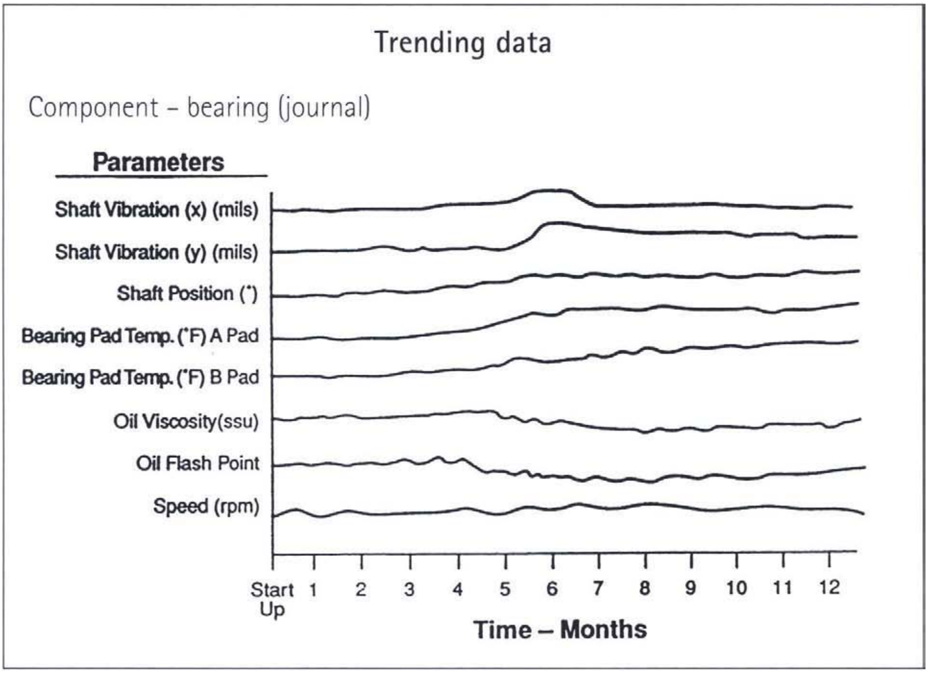


Figure 4.5 Trending data

Specific machinery component and system monitoring parameters and their limits

On the following pages is contained information concerning what parameters should be monitored for each major machinery component to determine its condition. In addition, typical limits are noted for each component.

These limits represent the approximate point at which action should be planned for maintenance. They are not intended to define shutdown values.

The rotor

Rotor condition defines the performance condition (energy and efficiency) of the machine. Figure 4.6 presents this value for a pump.

Pump performance monitoring

1. Take value at minimum flow (shut off discharge valve)
2. Measure:
   - P_1
   - P_2
   - Driver bhp
   - Specific gravity

   Where: P_1 and P_2 = psig, bhp = brake horsepower.
3. Calculate:

   A. head produced $\frac{\text{ft-lb}_f}{\text{lbm}} = \frac{\Delta P \times 2.311}{\text{S.G.}}$

   B. pump efficiency (%) $= \frac{\text{hd} \times \text{gpm} \times \text{SG}}{3960 \times \text{bhp}}$
4. Compare to previous value if > −10% perform maintenance

Figure 4.6 Pump performance monitoring

Radial bearings

Figures 4.7 and 4.8 present the facts concerning anti-friction and hydrodynamic (sleeve) radial or journal bearing condition monitoring.

Condition monitoring parameters and their alarm limits

Journal bearing (anti-friction)

Parameter	limits
1. Bearing housing vibration (peak)	.4 inch/sec (10 mm/sec)
2. Bearing housing temperature	180°F (85°C)
3. Lube oil viscosity	off spec 50%
4. Lube oil particle size	
■ non metallic	25 microns
■ metallic	any magnetic particle in the sump
5. Lube oil water content	below 200 ppm

Figure 4.7 Condition monitoring parameters and their alarm limits – Journal bearing (anti-friction)

Condition monitoring parameters and their alarm limits

Journal bearing (hydrodynamic)

Parameter	Limits
1. Radial vibration (peak to peak)	2.5 mils (60 microns)
2. Bearing pad temperature	220°F (108°C)
3. Radial shaft position*	>30° change and/or 30% position change
4. Lube oil supply temperature	140°F (60°C)
5. Lube oil drain temperature	190°F (90°C)
6. Lube oil viscosity	off spec 50%
7. Lube oil particle size	>25 microns
8. Lube oil water content	below 200 ppm

* Except for gearboxes where greater values are normal from unloaded to loaded

Figure 4.8 Condition monitoring parameters and their alarm limits – Journal bearing (hydrodynamic)

Thrust bearings

Figures 4.9 and 4.10 show condition parameters and their limits for anti-friction and hydrodynamic thrust bearings.

Condition monitoring parameters and their alarm limits

Thrust bearing (anti-friction)

Parameter	Limits
1. Bearing housing vibration (peak)	
■ radial	.4 in/sec (10 mm/sec)
■ axial	.3 in/sec (1 mm/sec)
2. Bearing housing temperature	185°F (85°C)
3. Lube oil viscosity	off spec 50%
4. Lube oil particle size	
■ non metallic	>25 microns
■ metallic	any magnetic particles with sump
5. Lube oil water content	below 200 ppm

Figure 4.9 Condition monitoring parameters and their alarm limits – Thrust bearing (anti-friction)

Condition monitoring parameters and their alarm limits

Thrust bearing (hydrodynamic)

Parameter	Limits
1. Axial displacement*	>15–20 mils (0.4–0.5 mm)
2. Thrust pad temperature	220°F (105°C)
3. Lube oil supply temperature	140°F (60°C)
4. Lube oil drain temperature	190°F (90°C)
5. Lube oil viscosity	off spec 50%
6. Lube oil particle size	>25 microns
7. Lube oil water content	below 200 ppm

* *and* thrust pad temperatures >220°F (105°C)

Figure 4.10 Condition monitoring parameters and their alarm limits – Thrust bearing (hydrodynamic)

Seals

Figure 4.11 presents condition parameters and their limits for a pump liquid mechanical seal.

Condition monitoring parameters and their alarm limits

Pump liquid mechanical seal

Parameter	Limits
1. Stuffing box pressure	<25 psig (175 kpa) **
2. Stuffing box temperature	Below boiling temperature for process liquid
3. Flush line temperature	+/- 20°F (10°C) from pump case temp
4. * Primary seal vent pressure (before orifice)	>10 psi (70 kpag)

* On tandem seal arrangements only

** Typical limit – there are exceptions (Sundyne Pumps)

Figure 4.11 Condition monitoring parameters and their alarm limits – Pump liquid mechanical seal

Auxiliary systems

Condition monitoring parameters and their alarm limits are defined in Figures 4.12 and 4.13 for lube and pump flush systems.

Condition monitoring parameters and their alarm limits

Lube oil systems

Parameters	Limits
1. Oil viscosity	off spec 50%
2. Lube oil water content	below 200 ppm
3. Auxiliary oil pump operating yes/no	operating
4. Bypass valve position (P.D. pumps)	change > 20%
5. Temperature control valve position	Closed, supply temperature > 130 55°C)
6. Filter ΔP	> 25 psid (170 kpag)
7. Lube oil supply valve position	change > +/-20%

Figure 4.12 Condition monitoring parameters and their alarm limits – Lube oil systems

Condition monitoring parameters and their alarm limits

Pump seal flush (single seal, flush from discharge)

Parameter	Limits
1. Flush line temperature	+/-20°F (+/-10°C) of pump case temperature
2. Seal chamber pressure	< 25 psi (175 kpa) above suction pressure

Figure 4.13 Condition monitoring parameters and their alarm limits - Pump seal flush

Figures 4.14, 4.15 and 4.16 present condition monitoring parameters and limits for dynamic compressor performance, liquid seals and seal oil systems. One final recommendation is presented in Figure 4.14.

Compressor performance condition monitoring

1. Calibrated: pressure and temperature gauges and flow meter
2. Know gas analysis and calculate k, z, m.w
3. Perform as close to rated speed and flow as possible
4. Relationships:

A. $$\frac{N-1}{N} = \frac{LN\,\frac{(T_2)}{(T_1)}}{LN\,\frac{(P_2)}{(P_1)}}$$

B. $$EFFICIENCY_{poly} = \frac{\frac{k-1}{k}}{\frac{n-1}{n}}$$

C. $$HEAD_{poly} = \left(\frac{Ft\text{-}lb_f}{Lb_m}\right) = \frac{1545}{MW} \times T_1 \times \frac{n}{n-1} \times Z_{aug} \times \left(\left(\frac{P_2}{P_1}\right)^{\frac{(n-1)}{n}} - 1\right)$$

5. Compare to previous value, if decreasing trend exists greater than 10%, inspect at first opportunity

Figure 4.14 Compressor performance condition monitoring

Condition monitoring parameters and their alarm limits

Compressor liquid seal

Parameter	Limits
1. Gas side seal oil/gas ΔP	
■ bushing	< 12 ft. (3.5m)
■ mechanical contact	< 20 psi (140 kpa)
2. Atmospheric bushing oil drain temperature	200°F (95°C)
3. Seal oil valve* position	> 25% position change
4. Gas side seal oil leakage	> 20 gpd per seal

* supply valve = + 25%
return valve = – 25%
Note this assumes compressor reference gas pressure stays constant

Figure 4.15 Condition monitoring parameters and their alarm limits – Compressor liquid seal

Condition monitoring parameters and their alarm limits

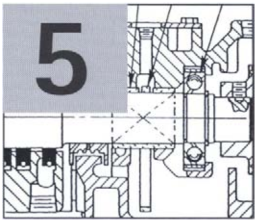
Compressor liquid seal oil systems

Parameters	Limits
1. Oil Viscosity	off spec 50%
2. Oil flash point	below 200°F (100°C)
3. Auxiliary oil pump operating yes/no	operating
4. Bypass valve position (P.D. Pumps)	change > 20%
5. Temperature control valve position	closed, supply temperature 130°F (55°C)
6. Filter ΔP	25 psid (170 kpag)
7. Seal oil valve position	change > 20% open (supply) > 20% closed (return)
8. Seal oil drainer condition	(Proper operation)
■ constant level (yes/no)	level should be observed
■ observed level (yes/no)	level should not be constant
■ time between drains	approximately 1 hour (depends on drainer volume)

Figure 4.16 Condition monitoring parameters and their alarm limits – Compressor liquid seal oil systems

Predictive maintenance (PDM) techniques

Now that the component condition monitoring parameters and their limits have been presented, predictive maintenance techniques must be used if typical condition limits are exceeded. The following chapter will address the techniques used for predictive maintenance analysis and root cause analysis techniques.



5

Effective predictive maintenance

(including root cause analysis techniques)

- Introduction
- Troubleshooting procedure overview
- Initial fact finding
- Thorough knowledge of equipment, component and system functions
- Defining abnormal conditions
- Listing all possible causes
- Eliminating causes not related to the problem
- State root causes of the problem
- Develop an action plan to eliminate root cause

Introduction

In this chapter, we will present the procedure used for both predictive maintenance and root cause analysis. You will find that many of the concepts covered in this section, and those required for effective troubleshooting have been previously mentioned and discussed. We have included exercises at the end of this chapter that will aid in the implementation of the principles discussed.

What is troubleshooting?

Troubleshooting is the action of discovering and eliminating causes of trouble. Figure 5.1 presents the definition.

Definition

Troubleshoot - To **discover** and **eliminate (root)** causes of trouble

Figure 5.1 Definition

Troubleshooting can be as simple as discovering why a light bulb does not function to as complicated as debugging modern day computer software. In this chapter we will deal with the important aspects of troubleshooting and practice those aspects to develop troubleshooting skills. Do not expect to acquire these skills immediately. Like any skills, they require practice. This chapter will introduce you to all of the important aspects and practice exercises. Reader's are encouraged to use the principles presented immediately and continue to implement these principles in order to perfect these skills. Figure 5.2 presents the basic requirements of troubleshooting.

- Troubleshooting requires that all **abnormal** conditions be defined
- However, to determine **abnormal** conditions, the **normal** conditions must be known
- Therefore **baseline (normal)** conditions must be known

Figure 5.2 The basic requirements of troubleshooting

We must first find abnormal conditions. To define abnormal conditions, the normal condition must be known. Therefore the baseline conditions are required. As mentioned in the beginning of this chapter the concept of normal condition and change of condition have been covered in the Chapter 4. We must be sure to define all abnormal conditions, consequently full condition monitoring must be practiced.

Predictive maintenance and troubleshooting

When the word troubleshooting is mentioned, most people immediately assume there has been some type of significant problem or failure. It is important to understand that the procedure of troubleshooting is exactly the same used in predictive maintenance. Figure 5.3 presents the similarity between predictive maintenance and troubleshooting.

Predictive maintenance and troubleshooting

- **Predictive maintenance** requires baseline and trend data to predict the root cause of the **change in condition**
- **Troubleshooting** requires baseline and trend data to predict the root cause of **failure**

Figure 5.3 Predictive maintenance and troubleshooting

The objective of any operating unit should be to practice predictive maintenance in order to eliminate the practice of troubleshooting. In this manner significant savings can be gained to the operating unit by eliminating unnecessary process unit downtime.

Obtaining and maintaining management support

As previously stated, management support must be obtained for any significant site activity. Especially in the case of troubleshooting when effective measures are required in a short period of time, management support is a must. The troubleshooting team must have the ability to ask required questions and use effective personnel during the initial phases of troubleshooting. Lack of management support will severely hamper this effort. Figure 5.4 presents the ways to obtain management support.

Obtaining and maintaining management support

Troubleshooting goes nowhere without continued management support

- Define impact of problem on profit
- Prepare brief outline of plan for management
- Include target dates

Figure 5.4 Obtaining and maintaining management support

Maintaining management support requires that the troubleshooting team provide management with frequent updates on the status of the troubleshooting effort.

Obtaining all the pertinent facts

In my experience, most troubleshooting exercises that fail do so because of incomplete facts or insufficient information. In the rush to define what the problem is many troubleshooting teams do not take sufficient time to obtain all of the facts. One of our team members once made the following statement when entering the refinery gates after a significant failure and speaking to the refinery manager. Note refinery was located (as most refineries) on a river. When the refinery manager asked what the problem was and what did he think the cause was, the representative promptly answered 'Harry, if I knew the answer to that question I would have walked across the river and not come in the gate'. We can see from the above statement that effective troubleshooting requires a thorough factual analysis. **Take your time**. Do not waste time, however, be complete and thorough in obtaining all of the facts. In addition, during the troubleshooting process it is usually necessary to go back and obtain more facts or test (substantiate) the facts already obtained. Figure 5.5 presents some ideas to consider when obtaining all the facts.

'Just all the facts please'

Most troubleshooting exercises that fail do so because of incomplete facts.

Consider the following:
- What has changed?
- Look at the total system
- Include all groups (maintenance, operation, engineering, contractor, vendor, etc.)
- Be informal (do not tape record!)

Figure 5.5 'Just all the facts please'

Thoroughly understanding the equipment and component functions

As mentioned, you will not become a troubleshooting expert in one week. The main reason for this fact is the knowledge required to define equipment function. This is precisely why many troubleshooters have gray hair. Seriously, effective troubleshooting must be based on a thorough knowledge of the equipment function. That is, what is the equipment supposed to do. In my experience, I have found it very effective to go one step further and divide the equipment into its various component and component systems. Then I ask the question what is the component and each of the components in its associated system supposed to do. It is obvious that a vast amount of knowledge is required to accomplish this fact. Remember, you may not have all of the knowledge required. Obtain it. Figure 5.6 presents sources of where the information may be obtained.

What is it supposed to do?

Thoroughly understand the function of each component by:

- Reading the instruction book
- Referring to reference books
- Asking questions of:
  - Peers
  - Vendors
  - Consultants

Figure 5.6 What is it supposed to do?

One final word. Do not be afraid to admit to management that you do not know certain aspects of the problem. But be sure to state that you will find out. After all, management must understand that troubleshooting is also a learning process and does require time.

To aid in the understanding of component function definition, we have included an example for a tilt pad bearing in Figure 5.7. As an exercise, state the required instruments that would have to be monitored to determine the bearing condition.

Component function example

A tilt pad bearing continuously supports all static and dynamic forces of a rotor by providing sufficient bearing area and oil flow to remove the generated frictional heat.

This statement then defines the parameters that must be monitored to determine the bearing's condition:

- Static and dynamic forces
- Bearing area
- Oil flow
- Frictional heat

Figure 5.7 Component function example

Defining effect, causes and root causes

Frequently troubleshooting confuses effect, cause and root cause. It is important that these three items are fully understood. Stop and think for a minute regarding troubleshooting problems that you have encountered, and ask the question, what is the first item identified? You will find on examination that problems are usually defined by their effects, the car won't start, the washer will not wash clothes, the television will not turn on. We can also examine typical machinery problems as follows: high vibration caused a shutdown, the bearing temperature high trip caused a shutdown, the extremely high seal leakage required a shutdown. All of these items are effects of the problem. Granted, they are causes of the shutdown, but remember, troubleshooting requires the effective definition of root causes for the problem. Figure 5.8 presents the differences between effects, causes and root cause. An exercise is included at the end of this chapter on determining the difference between effects, causes and root causes.

Effect, cause and root cause

- Problems are usually defined by effects
- An effect is a condition traceable to a cause
- A cause is the reason for a condition
- A root cause is the first cause in the chain of events

Figure 5.8 Effect, cause and root cause

The concept of system

I have found that after the problem of obtaining all the facts, the next most common pitfall in effective troubleshooting is failure to consider the system in which the component operates. Most root causes are in the system. Every major component that you will encounter is part of a system. Figure 5.9 presents these facts.

The concept of a system

- Each major equipment component is a part of a system
- Its condition is directly influenced by any system component's condition

Figure 5.9 The concept of a system

Always remember to first define the components involved and then thoroughly define the system in which the component acts. Frequently this system will be the process system, the lubrication system, the buffer gas system, etc. If it is an existing system, obtain that drawing (P&ID) and confirm the condition of each major component in that system. If it is not a common system (coupling system, etc.) sketch a schematic of that system noting each major component and again, confirm the condition of each component in that system.

Troubleshooting procedure overview

The intent of this subsection is to provide an overview concerning the troubleshooting procedure. In subsequent subsections specific details concerning each major item will be covered. Figure 5.10 presents the troubleshooting process logic diagram.

It is important that each step be thoroughly completed before proceeding to the next. Certainly there may be instances where it will be required to recycle and go back to a proceeding step to obtain more information or correct misinformation that exists. Remember this procedure is generic and may be slightly refined for specific problems. However, I have found this procedure to be the most consistent in effectively defining root causes and correcting those problems in a cost

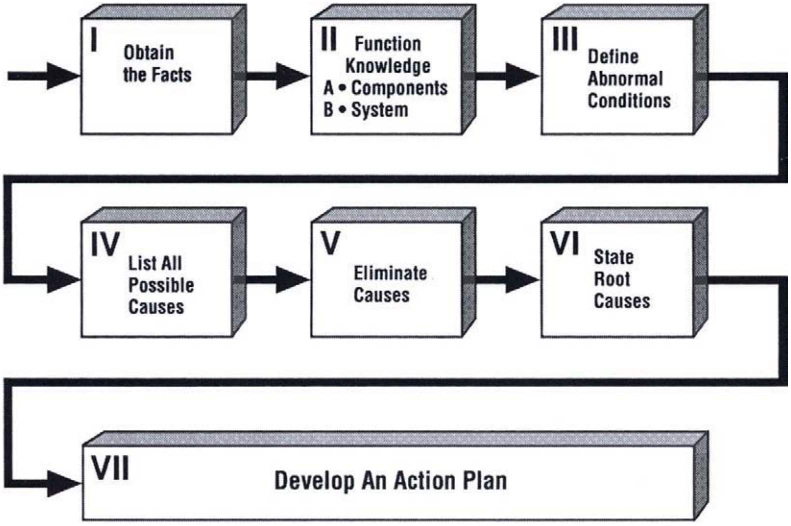

I Obtain the Facts
II Function Knowledge A • Components B • System
III Define Abnormal Conditions
IV List All Possible Causes
V Eliminate Causes
VI State Root Causes
VII Develop An Action Plan

Figure 5.10 The troubleshooting process

effective manner. Following are some thoughts that may be helpful in the overall troubleshooting procedure.

Initial fact finding

Do not leave any stone unturned. Ask all affected groups (operations, maintenance, engineering, contractor, vendor, etc.).

Thoroughly understanding affected component and system function

Do not be afraid to admit lack of knowledge. Obtain knowledge through experienced sources, instruction book and other publications. Confirm proper understanding of the facts before proceeding.

Defining abnormal conditions

Abnormal conditions may not mean that the condition is in alarm. It is helpful to use percentage to define deviation from baseline. Any significant deviation may define an abnormal condition.

Listing all possible causes

Note all causes, even if they appear to be highly improbable, causes can be eliminated when reviewing causes based on facts.

Eliminating causes based on facts

Review each cause listed in light of all facts. Eliminate causes that are not possible based on information.

Stating root cause(s)

In this area be sure to thoroughly investigate all systems and subsystems of components. Identify any subcomponents that are operating in abnormal conditions. Root causes are usually found in subsystems of components.

Developing an action plan

Include a concise action plan that can be presented to management to obtain full management support and aid in the implementation of the action plan.

Initial fact finding

As previously mentioned, this phase is the most important phase in the troubleshooting process.

Stating the apparent problem

Remember the apparent problem is usually an effect and not a cause. Figure 5.11 presents the major items involved in obtaining the apparent problem.

State the apparent problem

- Ask the involved personnel
- Inspect affected parts
- Confirm the apparent problem

Figure 5.11 State the apparent problem

Defining the affected components

Once the apparent problem is stated, all affected components must be defined. Figure 5.12 presents the action required to define these components.

Define all affected components

- Thoroughly inspect for all affected components
- List all affected components

Figure 5.12 Define all affected components

Obtaining all important facts

Once the problem and affected components are defined all the important facts surrounding these components must be known. Figure 5.13 is a suggested question list that has proven to be effective in fact finding. Remember, this is a generic list. Some questions may not be appropriate or additional questions may be required in specific cases.

Fact finding guidelines

Suggested question list

1. What is the problem?
2. What components failed?
3. What are facts concerning failed components?
4. How long has unit been operating without this component failure?
5. Has this component failed before?
6. What were component system parameters prior to failure?
7. What parameters exceeded normal values?
8. What changed?
   A. Process conditions
   B. Operating procedure
   C. New components (equipment and system)
   D. Piping system
   E. Foundation
9. Parts out of tolerance
10. Has this type of failure occurred in other locations?
    (Network - Users' groups)

Figure 5.13 Fact finding guidelines

Baseline conditions and trends

During fact finding activity it is extremely important to define all baseline conditions concerning the parameters around each major component involved. Refer back to the information in Chapter 4 regarding condition monitoring parameters and be sure that all of these parameters are checked for baseline condition and trends. This procedure may be very frustrating and may take a long time based on the data that the affected plant has available. Remember, be patient. Figure 5.14 lists the requirements to obtain baseline conditions and trends.

Baseline conditions and trends

- Establish conditions before failure (baseline)
- Utilize distributed control system, operator's logs and reliability data for baseline and trend data
- Establish changes from baseline conditions prior to failure
- Express condition changes in percent

Figure 5.14 Baseline conditions and trends

Inspection

Each affected component must be thoroughly inspected. This inspection will either add facts or possibly eliminate a component from consideration. Figure 5.15 presents information concerning component inspection.

Failed component inspection

- Thoroughly inspect all parts
- List all facts
- Utilize site experience
- Obtain vendor and/or consultant opinion if required
- Fully define inspection procedure and provide all facts if 'outside' inspection source is used

Figure 5.15 Failed component inspection

It may be necessary in this activity to enlist the help of additional associates as a check to inspection thoroughness. In addition, it may be necessary to call on outside sources (non-destructive testing companies, troubleshooting specialists, consultants, etc.). In this event, the effectiveness of their activity will significantly depend on how well their work effort is defined and how complete the information given to them is. Keep this in mind. Take the time required to thoroughly define what is required of outside sources and provide all of the information required.

Thorough knowledge of equipment, component and system functions

This area requires careful consideration. It also requires a significant amount of paperwork to first define the function of each component, define the system and list all of the effective parameters. Figure 5.16 presents these facts.

Component and system functions

- Define the function of each affected component
- Define the system in which each affected component operates
- List the normal parameters for each affected component and system component

Figure 5.16 Component and system functions

This activity will, most of the time, require confirmation of component function from instruction book sources, outside material, articles etc. or discussion with equipment experts. Be sure that the function of each component is properly defined in a simple manner before proceeding, but the definition must be complete. Once the component function is defined, then list all systems in which the affected component operates. Next list each parameter for the affected component that must be checked for condition. Again, the information contained in Chapter 4 concerning component parameters will be helpful.

Defining abnormal conditions

Once all component conditions are listed obtain information concerning baseline and trends. Figure 5.17 presents these facts. Again, it is cautioned that an abnormal condition may not necessarily be an alarmed condition. This could occur for many reasons; improper instrument setting, improper instrumentation functioning, high setting, etc. It is helpful to list in percentage the deviation between normal (baseline) and abnormal conditions. It may be necessary to consult other experienced sources regarding a certain deviation to determine if in fact, this abnormality is significant.

Define abnormal conditions

- Note value of each abnormal condition based on facts
- Express abnormal conditions in percent above normal limits
- State when each abnormal condition appeared

Figure 5.17 Define abnormal conditions

Listing all possible causes

Figure 5.18 presents information concerning causes.

Listing all possible causes

- Start with most obvious causes
- Obtain additional information if necessary
- List remaining causes
- Sub divide into possible cause sources

Figure 5.18 Listing all possible causes

It has been my experience that in many instances people start with very complicated causes and causes that are difficult to prove. Start with the most obvious causes from the fact finding phase and obtain more facts if necessary in order to list a cause. Again, be thorough and take your time. Insufficient information is the cause of most troubleshooting activity failures.

Once the possible causes are noted the sources for these causes must be defined. Subdivide the causes into the possible cause sources. The major causes of machine failure, as previously defined in Chapter 2 are:

- The effects of the process
- Assembly/Installation
- Operating procedures
- Design and manufacturing deficiencies
- Component wear out.

Use the facts obtained to determine if the causes qualify for the above sources. As an example, a design problem usually shows up from the first day of operation. If the problem has recently occurred, design and manufacturing can be eliminated (if a new part was not recently installed). In this case, the operating procedures and process conditions should be checked.

Eliminating causes not related to the problem

Refer to Figure 5.19. Review each cause noted and test this cause against all of the facts. Again, in this section it may be necessary to recycle and obtain additional facts or confirm facts previously stated.

Eliminate causes
not completely supported by facts

Figure 5.19 Eliminate causes

State root causes of the problem

Figure 5.20 presents the facts concerning root causes. It is important to mention in this section that the thorough definition of component systems and subsystems must be completed. If the affected component systems and subsystems are not thoroughly defined important information regarding root causes may be neglected. Once the systems are defined all of the components in those systems must be investigated for abnormalities.

State root cause(s)

- Consider cause sources
- Obtain additional input if necessary
- Be sure all component systems and sub systems are considered

Figure 5.20 State root cause(s)

Develop an action plan to eliminate root cause

Figure 5.21 presents guidelines in developing an action plan.

Develop an action plan

- Clearly state root cause
- Define revenue loss if problem is not corrected
- Clearly state action plan and define responsibilities
- Present plan to management

Figure 5.21 Develop an action plan

It is important to develop the plan in an outline fashion that can be effectively presented to management. The method in which the action plan is defined, written and presented will have a significant effect on whether the action plan is implemented. Be sure to emphasize the impact on profit to management in presentation of the plan.

Before an action plan is presented it may be necessary to hold meetings with contractors, vendors and/or consultants to thoroughly define all action required. The more complete an action plan, the better the chances for its success.

Exercise 1

Title: Fact finding

Purpose: Demonstrate fact finding procedure

Task: List all facts obtained from case history and questions to plant personnel. List all facts using the outlined procedure.

Given: Case history:
A single stage steam turbine driving a main lube oil pump in a NGL unit process centrifugal compressor tripped. The auxiliary pump did not start in time causing the compressor to trip on low lube oil pressure.

Exercise 2

Title: Component function exercise

Purpose: To demonstrate how a concise component function statement can define required component condition parameters

Task: Complete exercise by:

1. Stating component function
2. Circle key component condition parameters
3. Defining all instruments and/or components that must be condition monitored

Complete this exercise for the following components:

A. Radial bearing
B. Pump shaft seal
C. Single stage steam turbine shaft seal
D. Governor system for a single stage turbine
E. Pump rotor

1. Component function:

A. ____

____

____

____

2. Circle key component condition parameters

____

____

____

3. Define all instruments and/or components that must be condition monitored

____________________ ____________________

____________________ ____________________

____________________ ____________________

Exercise 3

Title: Component system exercise

Purpose: To enable the reader to define an entire component system

Task: Complete exercise by defining all the systems and system components that affect the condition of the noted component

Given:
A. Radial bearings
B. Balance drum
C. Sump pump rotor
D. Pump mechanical shaft seal
E. Single stage steam turbine shaft (floating carbon ring) seal

1. Component

____

____

____

2. Define all systems affecting the component in item 1

____

____

____

3. Define the major components in each system

______________ ______________

______________ ______________

______________ ______________

Exercise 4

Title: Effect, cause and root cause exercise

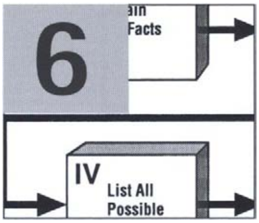
Purpose: To enable the reader to define cause, effects and root causes of problems

Task: Review the attached fact list and separate each fact into its appropriate category

Given:
A. High vibration pump inboard radial bearing
B. Pump cavitation
C. Plugged pump suction screen
D. Low (almost zero pump) discharge pressure
E. Failed mechanical seal
F. Low seal housing pressure
G. Plugged seal flush line
H. Failed radial bearing
I. Single stage steam turbine trips on start-up
J. Increased balance line ΔP

Review fact list and put fact in appropriate column:

Effect	Cause	Root cause
____________	____________	____________
____________	____________	____________
____________	____________	____________
____________	____________	____________



6

Root cause analysis example problem

- Introduction
- Example case history
- Answers and comments for the example case history

Introduction

In the previous chapter, the root cause analysis procedure was defined and discussed. In this chapter, an example is presented that will enable you to implement the procedure. The example is arranged to allow you to perform each step and then either immediately review the comments for that section (Considerations) or to proceed to the next step of the example. It is suggested that you attempt to complete the entire procedure the first time and then review the specific comments for each section. Answers for each section are contained in the back of this chapter.

The procedure outline from Chapter 5 is presented in Figure 6.1. Be sure to follow each step in order and to refer to the details for each step in Chapter 5 to assure completeness.

Example case history

The example case history is noted below. This is an actual problem that was encountered and resulted in multiple shutdowns of a process unit and loss of MM$ of profit.

Read the case history carefully and be sure to review all details. Also make a list of the questions that you would want to ask and to what

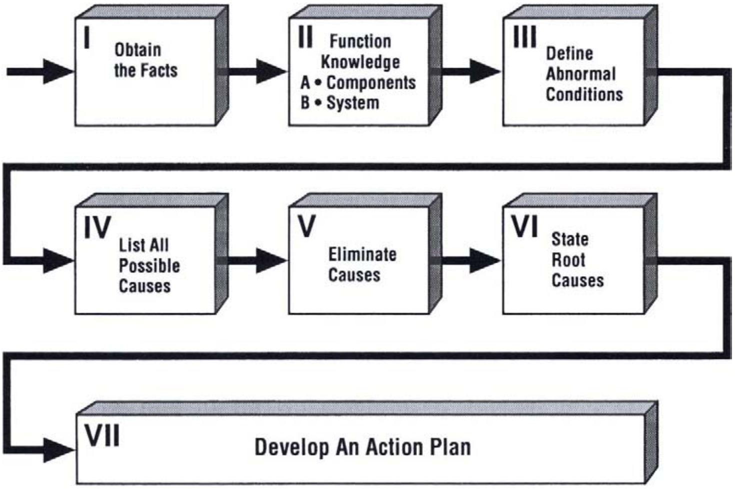

I Obtain the Facts
II Function Knowledge A • Components B • System
III Define Abnormal Conditions
IV List All Possible Causes
V Eliminate Causes
VI State Root Causes
VII Develop An Action Plan

Figure 6.1 The troubleshooting process

group in the plant (Operations, maintenance and engineering) you would direct your questions.

Title: **Example troubleshooting problem**

Purpose: To learn that troubleshooting should follow through logical steps, and requires knowledge of equipment and supporting system function and *practice*.

Task: Read the following case history and list any required questions. Follow the troubleshooting guidelines previously discussed. Complete *all* required steps, note *all* information required and state:

- Root cause
- Action plan

Given: Case history:

A two (2) stage motor driven positive displacement diaphragm compressor and an identical spare were installed in a chemical plant. Specifics are as follows:

I. Mechanical details (Figure 1)

A. Motor driven through a belt
B. 400 rpm compressor speed
C. Sleeve bearings – pressure lube
D. Self contained oil system in sump
E. No auxiliary pump (Manufacturer's proven design that had good field operating experience)
F. Main pump driven off the shaft

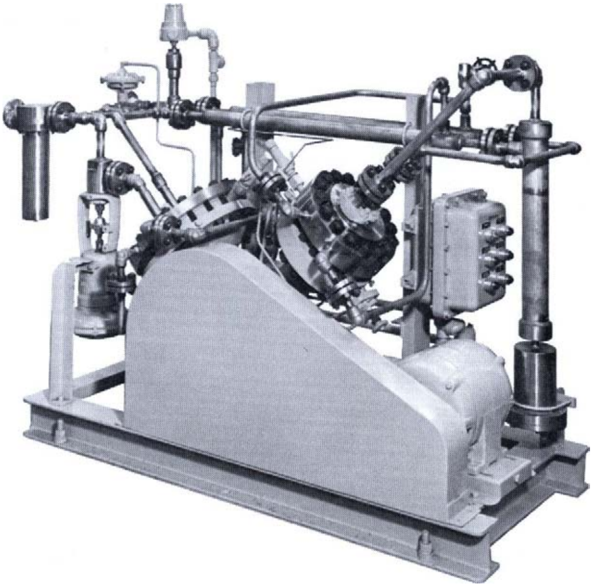


Figure 6.2

II. Performance details

P_1 = 80 psia
P_2 (final) = 330–400 psia
Intercooled
T_1 = 100ºF
T_2 (final) = 275ºF
Gas = dry nitrogen
Acfm (inlet) = 20 ft.^{3}/minute

III. Process system details (Figure 2)

The subject compressor supplies N_2 at approximately 300 psi, to catalyst feeder, to allow dry (non-stick) feed of catalyst to the reactor. Feed is continuous. Design of system incorporates a receiver downstream of compressor to hold excess quantity of N_2. Low and high pressure switch turn compressor on and off as required. The volume of the receiver, is 200 ft.^{3}. Approximate required feed rate to catalyst feeders = 1 acfm.

Pressure switches set at:

- Start compressor 300 psig
- Stop compressor 355 psig

IV. Additional facts

A. Unit was started up new.
B. Compressor ran well for first two weeks.

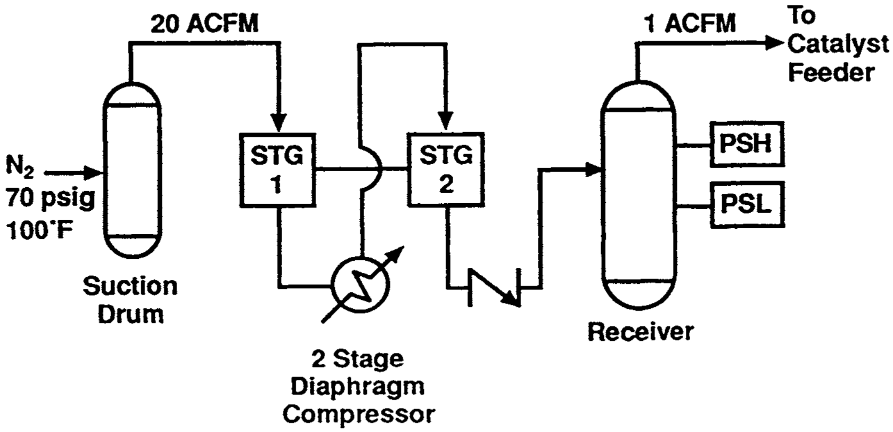

20 ACFM
1 ACFM
To Catalyst Feeder
N2
70 psig
100°F
Suction Drum
STG 1
STG 2
PSH
PSL
Receiver
2 Stage Diaphragm Compressor

Figure 6.3

C. After this time, both compressors continuously suffered bearing failures.
D. Bearings were totally black.
E. Babbitt was not found on bearing shells.
F. All clearances on bearings ok when installed.
G. Babbitt material per specification.
H. No abnormal vibration prior to failure.
I. All pressures and temperatures OK. Note: since this is a small unit, it is not supplied with vibration and bearing temperature monitoring.

Step 1 – Fact finding

This step forms the foundation for the procedure and determines its effectiveness.

Apparent problem: ______________________________

Affected components: ______________________________

____

____

Facts: ____________________ ____________________

____________________ ____________________

____________________ ____________________

____________________ ____________________

____________________ ____________________

Additional questions:

____

____

____

____

Considerations:

Be sure to:

- First determine the apparent problem from the facts presented. What is it? It may be necessary to change the apparent problem during the procedure based on the information gathered.
- Next, determine the affected components(s). What are they?
- If the information presented in the case history requires questions, ask them of the appropriate plant group (s). Asking the same question of different groups can be helpful.
- Review the apparent affected components(s) damage details thoroughly.
- Now obtain all the facts. Use the failure classification list contained in Chapter 2 as a checklist to assure that questions are asked concerning all the failure classifications. Also refer to the Fact list contained in Chapter 5.
- In addition, a good example of questions to ask is contained in the site study questionnaire contained at the back of Chapter 8.

Step 2 – Component and system function knowledge

This step requires the most work and the greatest knowledge base. Next to insufficient fact gathering, it is the area where most root cause analysis procedures fail. In fact, this step is so important that the next chapter is devoted to increasing this skill. And, no matter what your level of experience, this task will never be complete. There is always more information to learn and store in your data base.

Failed component(s) function – what is it supposed to do?

____

____

____

____

Systems that support the failed component

Components in each system that support the failed component

System: **Components:**

Parameters for input data:

Possible failure causes for each affected component and system:

Considerations:

- First define the component function definition in clear, concise terms for each affected component noted in the previous step.
- Next, define the systems and subsystems that support each component.
- Then define all of the parameters necessary to obtain data for each component and all of its systems and subsystems.

- Be sure to utilize the component condition monitoring lists presented in Chapter 4 when obtaining information.
- Now obtain this information first from DCS and PLC systems if possible before asking questions or using manual logs to assure accuracy. However, pay attention to the information and assure that the sampling rates are sufficient for the trends that you require.
- Once this information is obtained, start to think of the possible failure causes, based on your present knowledge base, and list them for each component and it's systems and sub-systems.
- Do not hesitate to consult reference books, instruction books and to question associates both in your plant, other plants in your company, Internet and Network sources (from past conference, training workshops etc.)

Step 3 – Defining abnormal conditions

Use the data obtained at this point to define abnormal conditions. Do not look for only alarms!! If baseline data (data before the event) was low, a significant change in the parameter value could have occurred without alarm actuation!

Abnormal condition parameters:

____

____

____

____

Questions and group to confirm each abnormal condition:

____

____

____

____

Considerations:

- Use % change to define abnormal conditions.
- I have used a 50% value as a norm but, it depends on the application and the signature of the specific machine.
- List all abnormal conditions that exceed your defined % value.
- Review the failed component function definitions, systems and sub-systems and prioritize the abnormal conditions ...

- List additional questions for each plant group to confirm that each noted abnormal condition is valid based the history of this machine.

Step 4 – Listing causes

Now, list all possible causes, based on your expanded knowledge base and information gathered to date.

Possible causes:

____

____

____

____

Additional questions to confirm causes:

____

____

____

____

Considerations:

- Start with the simple causes first.
- Be sure that only causes are listed and not effects.
- Confirm causes by reviewing data an asking additional questions.
- Consult associates in other locations and/or use company network to confirm the validity of the causes.

Step 5 – Eliminating causes

Once the causes are noted, screen each cause based on all the information gathered. Eliminate each cause not totally supported by the facts.

Causes eliminated	__________	**Reason**	_______________
	__________		_______________
	__________		_______________
	__________		_______________

Considerations:

- Carefully screen each listed cause and eliminate those not fully supported by facts.

Step 6 – Determining the root cause

State the root cause.

Root cause(s)

____

____

____

____

Considerations:

- Dig deep.
- Most root causes are found in the systems and subsystems that support the failed component(s).
- Root causes frequently involve lack of awareness of personnel.

Step 7 – Constructing an action plan

List the proposed action plans:

____

____

____

____

Considerations:

- Justify the action plan to management based on lost profit.
- List the cost of the action required and show savings as a result.
- Make sure the action plan is detailed with responsible persons listed for each task.
- Follow up on all action points to be sure they are implemented.
- If contractor or vendor design meetings are required, prepare a detailed agenda, listing required individual disciplines and be sure to forward it well in advance of the meeting.
- Hold design meetings at the vendor's offices to be sure all required specialists are available.

Answers and comments for the example case history

Answers are arranged in the order of the procedure and include comments for each section of the problem.

1. Facts

The apparent problem – *all journal bearings failed*

Comments:

The apparent problem should always be an effect, not a cause. The tendency is to jump immediately to a cause of the problem. Even if the cause or the root cause is apparent to you, stick with the procedure format. Many a root cause analysis has produced results that are not complete which leads to another root cause analysis after another failure. Note also that the apparent problem has to be precisely stated – all journals bearings failed, not just one of the bearings.

The affected component(s) – *sleeve type journals bearings*

Comments:

Be sure to be precise in listing the failed component. An exact definition is necessary to determine the required parameters to check for causes and root causes. As an example, there are many types of journal bearings and each type has different characteristics, supporting systems and failure causes. Using the incorrect type could lead the analysis in the wrong direction and draw incorrect conclusions. For the present example, noting only a bearing would leave the analysis open to Anti-friction (ball, roller or tapered roller), sleeve, multilobe, tilt pad etc. bearings and their many types of lubrication systems (ring oil, oil mist, pressurized etc.). Be aware that the normal tendency is to think along the lines of your experience. Consult experienced associates, publications and the internet for information concerning the specific affected component.

The facts: failure class number is noted by ( )

- Pressurized lubrication system (1)
- No auxiliary lube oil pump (1)
- Main pump is shaft driven (1)
- The compressor is stopped and started by pressure (volume remaining) in the receiver vessel (1)
- The compressor should not stop and start often based on design values (200 cubic feet receiver capacity and 1 cubic foot per minute flow required) (1)
- There were no problems during the first two weeks of operation (3)

- Condition of all bearings (totally black) showed indication of lack of lubrication and/or excessive load. (1)
- It was confirmed that bearings were correctly installed (2)
- Bearing material was per specification (4)
- Measured vibration and compressor frame bearing housing temperatures using portable instruments were ok (4)

Additional questions (ask additional questions, based on the facts noted and note the discipline required).

- What changed after 2 weeks of operation? (Operations, maintenance instrumentation, process engineers) – the only change was the amount of flow required by the catalyst feeder. It was 1 cubic foot per minute but was changed to 10 cubic feet per minute by the process design company
- What was the design stop and start time of the compressor? – Approximately 6 times a day
- What was the actual stop and start time observed? – Approximately one every 5 minutes
- Have other plants using the same technology had the same problem to date? – No
- Has the compressor vendor experienced this type of problem in the past? – No

Comments:

Use the failure classification in recording the facts:

1. The effect of the process and associated systems.
2. Improper assembly/disassembly.
3. Improper operating procedures.
4. Design and manufacturing deficiencies.
5. Component wear out.

It has been my experience that the majority of root causes are found in failure classification 1. This is also the most difficult failure classification to define since it requires detailed knowledge of the process and associated systems.

Do not limit the input information to what you obtain, involve all required disciplines in the plant.

Refer to the fact list in Chapter 5 and be sure to ask these important questions. Especially, has it always been a problem and what changed?

Be sure to inspect all damaged components very carefully.

Be specific regarding the question what has changed and be sure to ask regarding each component in the supporting systems for the failed component.

2. Failed component function

A sleeve journal bearing supports the load with an oil film of the correct thickness and the proper viscosity to reduce friction.

Components present in the system that supports the failed component.

- Oil sump in the base of the compressor.
- Oil strainer in the line to the main pump.
- Main pump.
- Coupling between the pump and the shaft.
- Oil filter (no oil cooler required based on the specified stop and start time).
- Pressure control valve.

Parameters for input data.

- Oil sample analysis, including particle analysis.
- Measured bearing housing temperatures.
- Measured vibration on bearing housing.

Possible failure causes for each affected component and system:

- Lack of lubrication.
- Insufficient bearing clearance.
- Excessive bearing clearance.
- Excessive bearing load.
- Internal misalignment between bearings.
- Improper bearing Babbitt material specification.

Questions:

- Was the proper oil used? – Yes
- Was the oil type changed after the first 2 weeks? – No
- Was there any wear on the oil system components? – No
- Was there any blockage of the suction strainer or the filter? – No

Comments:

- Note the answers above carefully and compare to the facts stated in section 1 of the procedure.
- Knowledge of not only the failed component and its systems is necessary but also knowledge of the design characteristics of the failed machine. In this case, a positive displacement compressor has the characteristic of constant volume flow for a given speed.

3. Abnormal conditions

- The number of stops and starts per unit time.
- The lack of the ability to remove heat and lubricate all journal bearings.

Comments:

- Abnormal conditions are usually defined using trends of instruments. However, on small machines such as this, little if any data is trended and the team must rely on either observed facts (the best source) or information from other sources. The number of stops and starts was observed during operation after the failures had begun.

4. List the possible causes based on facts and abnormal conditions

- Poor lube system design – no auxiliary pump.
- Poor control system design – compressor is stopped and started.
- Change of the required flow – increase of 10 times.
- Rapid stops and starts – every 5 minutes therefore not providing proper lubrication to the bearings.

Comments:

- Be sure to look for the simplest causes first.
- Screen listed cause to be sure that they are not effects.
- If there are possible effects listed, it may be a cause but not a root cause.

5. Eliminate causes

Causes eliminated (causes eliminated with reasons for)

- Poor lubrication system design – proven vendor experience and no problems experienced during the first two weeks of operation
- Poor control system design – this system was used in all other plants of the same design with no reported problems to date.

Comments:

- The action taken above required detailed discussion with the compressor vendor, process designer and users in other plants.
- It is important to network with other plants in your company or user groups in your geographical area.
- Attending industry conferences and recommended training sessions by associates is also a source of increased knowledge base that can optimize your effectiveness.

6. State the root causes remaining

- Change of process conditions – required flow increase by 10 times.
- Rapid stops and starts of the compressor.

Root cause – change of process conditions

Comments:

- Since the type of compressor in this case is positive displacement, it is important to know that the volume flow rate of 20 cubic feet per minute is constant for a given speed. Note that this application is a fixed speed belt driven compressor.
- Rapid stops and starts are related to change of the required flow. Since the flow produced by the compressor is constant, the increased flow now required by the process will reduce the pressure in the receiver vessel more rapidly and result in more frequent stops and starts.
- Therefore, the change of the required flow was the root cause.
- It could be argued that the root cause was the process designer's lack of awareness of the effect on a positive displacement compressor, on stop/start control, when required system flow is significantly increased.
- The final task is to define a simple, cost effective action plan that can be rapidly implemented.

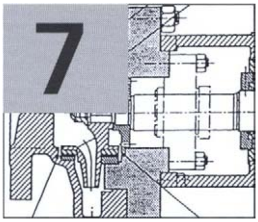
7. Action plan

- Confirm that the requirement for a 10 fold increase of flow is a real requirement.
- Eliminate the stops and starts on the compressor by disconnecting the pressure switches from the motor starter and operate the compressor continuously.
- Install a bypass control valve from the receiver to recycle back to the suction of the compressor to maintain a constant receiver pressure of 350 psig.

- Switch over the compressors for diaphragm inspection (initially every 4 months based on experience) and extend the inspection time based on observed findings.

Comments:

- A meeting was held with the process design company to confirm that the requirement for the increased flow was justified and resulted in increased process unit reliability and revenue. This fact was confirmed in the meeting.
- Present the action plan to management based on the loss of revenue if the problem is not corrected. Be sure to include the cost and time required for the modification. The final action plan should be a proven, cost effective approach with the shortest implementation time.
- Note that in this case, a blank flange for the bypass connection existed on the receiver vessel. The required valve was immediately available and the compressor intercooler capacity was checked and confirmed to be acceptable for the revised case.
- The option to increase the compressor capacity to minimize the number of starts, was not implemented due to the required modifications to the pulley system (to increase the speed and therefore capacity), the need to install a new motor and the lack of compressor experience at the increased speed.



7

Root cause analysis techniques

(Improving component function knowledge base)

- Introduction
- Component function
- Component failure causes
- Component condition monitoring
- Examples of knowledge base enhancement

Introduction

The purpose of this chapter is to present important information concerning component and supporting system function, failure causes and condition monitoring requirements to increase your knowledge base for use in the root cause analysis procedure.

A sound component knowledge base is an essential tool for use in effective predictive maintenance and root cause analysis programs. Next to obtaining all the pertinent facts, knowledge base is a corner stone in the root cause analysis process. No matter what your level of knowledge, it can be improved. And, with process design improvements being made frequently, machinery design requirements and component functions are changing so that continuous improvement is a necessity.

A recent root cause analysis that I was connected with is an example. The apparent failure was a steam turbine rotor severe rub to the inner casing of a high pressure condensing turbine application. Lack of specific knowledge, on my part, regarding the thermal expansion characteristics of the inner casing was leading to false conclusions in the RCA process. I requested a design audit meeting with the machinery supplier and reviewed the finite element analysis (FEA) of the inner casing. Once that knowledge or awareness of the design basis was obtained, the possible causes of the failure were easily identified.

Figure 7.1 presents this important fact.

Component function

'If you don't know *what* the component is supposed to do and *how* it does it, effective analysis is impossible'

Figure 7.1 Component function

As previously stated in Chapter 4, regardless of the type of machinery, there are five (5) major components and systems:

- The rotor
- The bearing(s) – radial
- The thrust bearing
- The seal(s)
- The auxiliary systems

It is assumed that the reader has previous experience with rotating equipment. Most likely, it will be experience with the use, assembly, disassembly, predictive or preventive maintenance of the machinery. Unless you have worked for a machinery supplier at one point in your career, your knowledge base regarding the design basis of the components and their supporting systems will be limited.

Therefore, the objective of this Chapter is to 'stretch' and 'improve' your component function knowledge base for those machinery components that typically have the lowest reliability and availability.

The important components and systems that are present in all types of machinery are shown in Figure 7.2.

Machinery component/system knowledge base examples

1. Centrifugal rotors – effect of the process and fluid head concepts
2. Radial bearing(s)
3. Thrust bearings and balance drums
4. Seals
5. Auxiliary systems – lubrication, seal, buffer and cooling

Figure 7.2 Machinery component/system knowledge base examples

The method used to determine your present component knowledge base and expand it is outlined in Figure 7.3. Information concerning the design basis for each of the components and systems noted in Figure 7.2 is included in this chapter.

Component knowledge base enhancement procedure

For each selected component:

I. Answer the following questions:
   1. Component function – 'What it does'
   2. Component operation – 'How it does it'
   3. List failure causes for the component
   4. List component monitoring parameter required to detect component condition changes (refer to Chapter 4)

II. Then review the reference information included in this chapter and modify the answers in item I as required.

III. Continue to modify the answers in item I as you acquire additional knowledge and keep the information handy for use in all RCA procedures.

Figure 7.3 Component knowledge base enhancement procedure

For each of the components and systems noted in Figure 7.2, you will be asked to complete the following activities:

1. Component function exercise
2. Component failure cause exercise
3. Component monitoring parameter exercise

You are encouraged to make time to complete these exercises for each of the five items contained in this chapter. Keep in mind that every machinery type you have or will ever encounter contains these components and systems! Having a sound knowledge base of component and system function, failure causes and monitoring parameters will greatly improve your RCA skills.

These exercises should be scheduled over time but completed as soon as possible. Once the information is listed, keep it available and update it after very RCA, article you have read, conference you have attended, supplier technical presentation etc. Your machinery component knowledge base is really the foundation of your troubleshooting skills! And the foundation must be checked and fortified periodically to support your efforts. This will be an ongoing process and should never stop as long as you are associated with rotating machinery predictive maintenance and problem analysis.

Regardless of your experience level, it is suggested that you complete these exercises for each of the five items, before reading the supporting information contained in this chapter or before obtaining information from other sources.

Then review the information carefully and make any required changes. It is suggested that you then concentrate on one item at a time and read the information contained in this chapter and modify your initial answers accordingly. After you have completed this step, discuss your answers with associates if possible and be sure to include all available disciplines when discussing these items operations, maintenance process engineering, reliability engineering etc.). Finally, consult supplier instruction books, available books on the subject, magazine articles and of course the web for additional information. Update your information and continue to do so.

Actually, this process is no different than the action the machinery suppliers take! They have an established experience data base and continually update it with new information to produce a maximum data base of correct information that will enable them to design new equipment and solve reliability issues. You may ask, why then do I need to obtain this information? My answer is ... because your company's objective and the suppliers objective are the same, to maximize profit, but the methods to achieve that objective are directly opposed! Your company maximizes profits by operating the unspared (critical) machinery in the plant 24/7 and identifying potential reliability issues through effective predictive maintenance to meet that objective.

The suppliers maximize profit by manufacturing the machinery to meet your specifications at the lowest cost to assure that the equipment will be reliable for the warrantee period of the equipment. Certainly, this is not to say that planned obsolescence is a supplier design objective, but suppliers cannot stay competitive and in business if the equipment is designed for the life of the plant (30 years).

The reference information in this chapter constitutes the minimum that you will need to be an effective problem solver. Increase this information by reading the supplier instruction manuals in your area as a minimum.

Component function

Figure 7.4 requires you to define 'what' the component system or item is supposed to do and 'how' this task is accomplished. Complete this exercise for the selected component or system. It is suggested that you perform this exercise and the following failure cause and monitoring parameter exercises at one time for each selected component or system.

Function exercise

Component ____________ or system _________

I. 'What' the function of ____

_________ is to ____

____

____

____

II. 'How' the function of ____

_________ is accomplished ____________________________________

by ____

____

____

____

____

Figure 7.4 Function exercise

The following example in Figure 7.5 for a hydrodynamic thrust bearing, demonstrates the requirements of the exercise.

Function exercise

Component: *hydrodynamic thrust bearing*

I. 'What' The function of a *hydrodynamic thrust bearing* is to *support the rotor in the axial direction.*

II. 'How' The function of a *thrust bearing* is accomplished by *having sufficient bearing area to support the maximum anticipated thrust loads, be properly installed and be provided with clean, cool lubricating oil of the correct viscosity and flowrate.*

Figure 7.5 Function exercise

Component failure causes

Figure 7.6 requires you to list all of the failure causes you can think of for your assigned component or system. In the case of the thrust bearing example above, failure causes are:

1. Insufficient bearing area
2. More load than anticipated
3. Incorrect installation
4. Unclean oil
5. Hot oil
6. Incorrect oil viscosity used or incorrect oil temperature
7. Incorrect oil flowrate

As can be seen from the above example, the key is to have a concise function definition. Once that is achieved, the remainder of the exercise is relatively easy. Once the failure causes are defined, the monitoring parameters to review for trends can be defined.

Do not forget however that while the use of the function and supporting system definitions aid greatly in the effectiveness of the RCA procedure, all of the failure classifications noted in Chapter 2 must be considered. As an example, the cause of 'More load than anticipated' for the thrust bearing example above could be in 4 of the 5 failure classifications:

- The effects of the process – higher discharge pressure
- Assembly/disassembly errors – insufficient clearance
- Operation errors – closing the discharge valve

- Design (thrust balance) and/or manufacturing errors – supplier balance drum sizing error.

Component failure causes

Component ___________ or system _________

List all possible failure causes. (Note: examine each of the failure classifications – in preparing this list).

Reference
Number

1. ____
2. ____
3. ____
4. ____
5. ____
6. ____
7. ____

Figure 7.6 Component failure causes

For your reference, we have included the failure classifications in Figure 7.7.

Failure classifications

- Process condition changes
- Improper installation/maintenance assembly
- Improper operating procedure
- Design problems
- Component wearout

Figure 7.7 Failure classifications

Remember to use *all* of the knowledge available for these exercises! The information gathered for these exercises will give you a 'running start' in your RCA procedures whether they will be used for predictive maintenance analysis or root cause failure analysis. Once you have completed these exercises for each component and system consult associates in your plant for their input and go outside your immediate group. It is a fact that many RCA procedures stay in the discipline of the RCA leader. Most RCA exercises are confined to the maintenance department. The root causes of failure can and will be found in all areas of the plant and every discipline should be consulted and can contribute. Reliability (and root cause analysis) is everyone's responsibility!

Component condition monitoring

Figure 7.8 presents the monitoring parameter exercise. Complete this exercise for your selected component or system now. For each failure cause listed in Figure 7.6, note in the corresponding number reference in Figure 7.8 the parameters (instruments or instruments required to be used) to monitor to determine if that failure cause is present.

For the hydrodynamic thrust bearing example given in Figure 7.5, the monitoring parameters for each failure cause are:

1. More load than anticipated – Axial displacement, thrust pad temperature, oil supply and drain temperature, process conditions (flow, fluid composition (MW or SG), suction and discharge pressure), machine speed or guide vane angle
2. Insufficient bearing area – Axial displacement, thrust pad temperature, oil supply and drain temperature
3. Unclean oil – Oil particle analysis
4. Hot oil – Oil supply temperature
5. Incorrect oil viscosity used or incorrect oil temperature – Oil analysis and oil temperature
6. Incorrect oil flowrate – Install temporary ultrasonic flowmeter and possibly confirm by calculating flow thru supply oil valve using valve position, valve differential pressure, oil specific gravity and valve trim information (valve Cv). Also view thrust bearing sight glass for a flow indication.

The parameter monitoring points noted above were obtained from the component condition monitoring lists in Chapter 4.

Failure cause monitoring parameters

Component ___________ or system ________

List all parameters (instruments) to be monitored or used for each failure cause listed in Figure 7.6. (Note: use corresponding failure cause reference number (same reference number as in Figure 7.6).

Reference
Number

1. ____
2. ____
3. ____
4. ____
5. ____
6. ____
7. ____

Figure 7.8 Failure cause monitoring parameters

Once this information is obtained, condition monitoring trends to be reviewed can be easily identified and checked to define the causes of the component change of failure.

For the hydrodynamic thrust bearing example, axial displacement and thrust pad temperature trends may show when the thrust bearing load increased. Examination of process information system trends for the same time period (PI system, Aspen etc.) may show the specific cause of increased thrust bearing loading.

The remainder of this chapter contains the reference material for the major specific component or systems. It is strongly recommended that you read this information after you have first attempted to complete all exercises for each of the five major component and systems items mentioned in this chapter.

You are also encouraged to continuously improve your 'component knowledge' by:

- Obtaining articles
- Surfing the net
- Reading vendor publications

- Network with experienced associates
- Attend industry conferences
- Through additional site training courses (including supplier sponsored courses)

The answers for each of the five major components and systems in rotating machinery are contained at the end of this chapter.

Examples of knowledge base enhancement

In this section, I have included material that will provide a 'start' to your major component knowledge base for

- The rotor
- Thrust bearings
- Auxiliary systems
- Journal bearings
- Seals

The effect of the process on machinery reliability
(Rotor component knowledge)

- Introduction
- The major machinery components

Introduction

The effect of the process on machinery reliability is often neglected as a root cause of machinery failure. It is a fact that process condition changes can cause damage and/or failure to every major machinery component. For this discussion, the most common type of driven equipment – pumps will be used.

There are two (2) major classifications of pumps, positive displacement and kinetic, centrifugal types being the most common. A positive displacement pump is shown in Figure 7.9. A centrifugal pump is shown in Figure 7.10.

It is most important to remember that all driven equipment (pumps, compressors, fans, etc.) react to the process system requirements. They do only what the process requires. This fact is noted in Figure 7.11 for pumps.

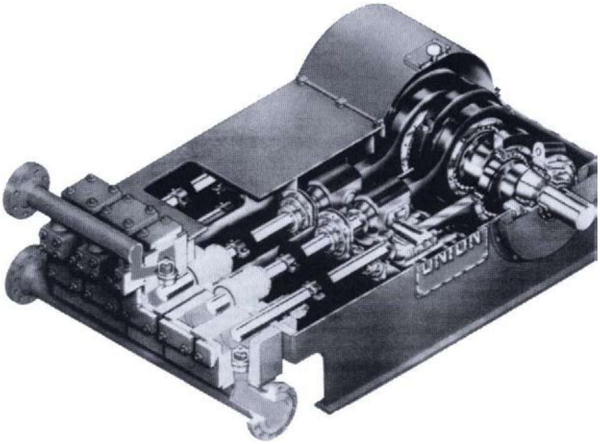


Figure 7.9 Positive displacement plunger pump (Courtesy of Union Pump Company)

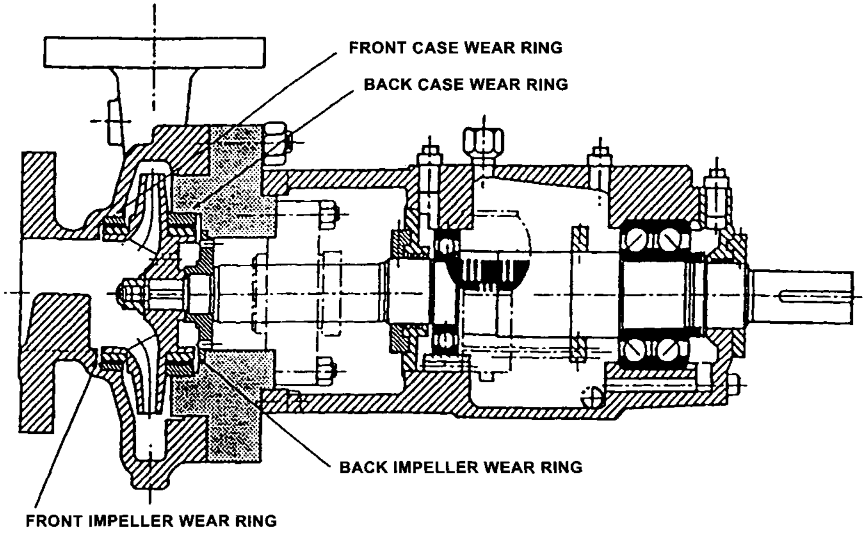


Figure 7.10 Centrifugal pump

Pump performance

- Pumps produce the pressure required by the process
- The flow rate for the required pressure is dependent on the pump's characteristics

Figure 7.11 Pump performance

Centrifugal (kinetic) pumps and their drivers

Centrifugal pumps increase the pressure of the liquid by using rotating blades to increase the velocity of a liquid and then reduce the velocity of the liquid in the volute. Refer again to Figure 7.10.

A good analogy to this procedure is a football (soccer) game. When the ball (liquid molecule) is kicked, the leg (vane) increases its velocity. When the goal tender (volute), hopefully, catches the ball, its velocity is significantly reduced and the pressure in the ball (molecule) is increased. If an instant replay 'freeze shot' picture is taken of the ball at this instant, the volume of the ball is reduced and the pressure is increased.

The characteristics of any centrifugal pump then are significantly different from positive displacement pumps and are noted in Figure 7.12.

Centrifugal pump characteristics

- Variable flow
- Fixed differential pressure produced *for a specific flow**
- Does not require a pressure limiting device
- Flow varies with differential pressure (P_1-P_2) and/or specific gravity

*assuring specific gravity is constant

Figure 7.12 Centrifugal pump characteristics

Refer again to Figure 7.11 and note that all pumps react to the process requirements.

Based on the characteristics of centrifugal pumps noted in Figure 7.12, the flow rate of all types of centrifugal pumps is affected by the Process System. This fact is shown in Figure 7.13.

Therefore, the flow rate of any centrifugal pump is affected by the process system. A typical process system with a centrifugal pump installed, is shown in Figure 7.14.

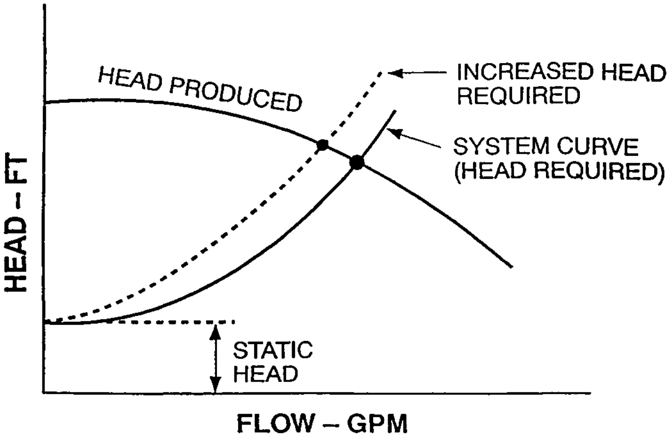


Figure 7.13 A centrifugal pump in a process system

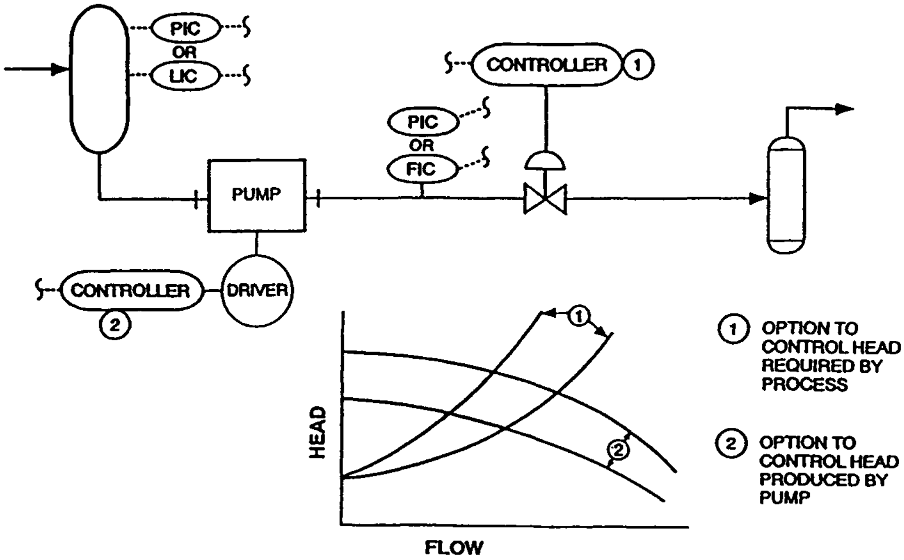

PIC
OR
LIC
CONTROLLER ①
PIC
OR
FIC
PUMP
CONTROLLER ②
DRIVER
HEAD
FLOW
① OPTION TO CONTROL HEAD REQUIRED BY PROCESS
② OPTION TO CONTROL HEAD PRODUCED BY PUMP

Figure 7.14 Centrifugal pump control options

The differential pressure required (proportional to head) by any process system is the result of the pressure and liquid level in the suction and discharge vessel and the system resistance (pressure drop) in the suction and discharge piping.

Therefore, the differential pressure required by the process can be changed by adjusting a control valve in the discharge line. Any of the following process variables (P.V.) shown in Figure 7.14, can be controlled:

- Level
- Pressure
- Flow

As shown in Figure 7.13, changing the head required by the process (differential pressure divided by specific gravity), will change the flow rate of any centrifugal pump!

Refer to Figure 7.15 and it can be observed that all types of mechanical failures can occur based on *where the pump is operating based on the process requirements.*

Since greater than 95% of the pumps used in this refinery are centrifugal, their operating flow will be affected by the process. Please

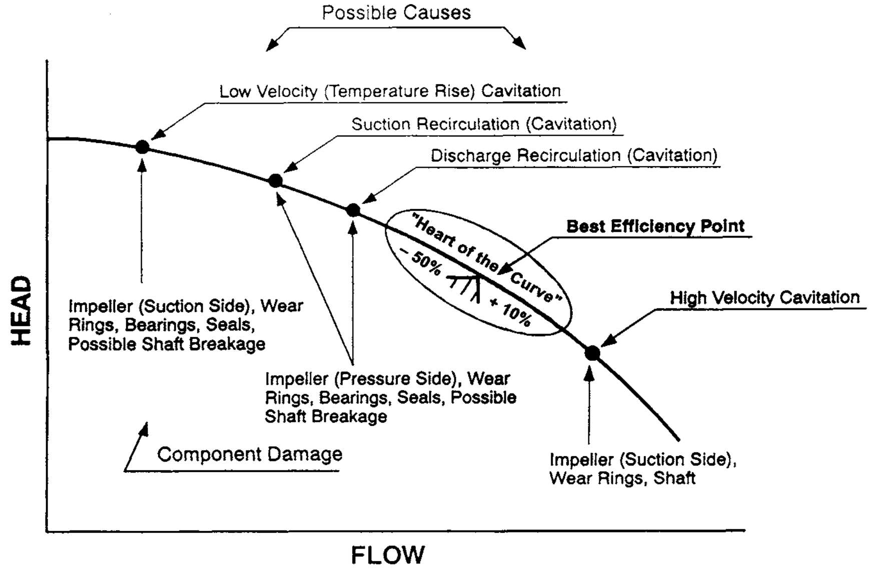

Possible Causes
Low Velocity (Temperature Rise) Cavitation
Suction Recirculation (Cavitation)
Discharge Recirculation (Cavitation)
Best Efficiency Point
"Heart of the Curve"
- 50%
+ 10%
High Velocity Cavitation
HEAD
Impeller (Suction Side), Wear Rings, Bearings, Seals, Possible Shaft Breakage
Impeller (Pressure Side), Wear Rings, Bearings, Seals, Possible Shaft Breakage
Component Damage
Impeller (Suction Side), Wear Rings, Shaft
FLOW

Figure 7.15 Centrifugal pump component damage and causes as a function of operating point

refer to Figure 7.16 which shows centrifugal pump reliability and flow rate is affected by process system changes.

Centrifugal pump reliability

- Is affected by process system changes (system resistance and S.G.)
- It is not affected by the operators!
- Increased differential pressure (P_2-P_1) means reduced flow rate
- Decreased differential pressure (P_2-P_1) means increased flow rate

Figure 7.16 Centrifugal pump reliability

At this point it should be easy to see how we can condition monitor the centrifugal pump operating point. Refer to Figure 7.17.

Centrifugal pump practical condition monitoring

- Monitor flow and check with reliability unit (RERU) for significant changes
- Flow can also be monitored by:
  - Control valve position
  - Motor amps
  - Steam turbine valve position

Figure 7.17 Centrifugal pump practical condition monitoring

Driver reliability (motors, steam turbine and diesel engines) can also be affected by the process when centrifugal driven equipment (pumps, compressor and fans) are used.

Refer to Figure 7.18 and observe a typical centrifugal pump curve.

Since the flow rate will be determined by the process requirements, the power (BHP) required by the driver will also be affected. What would occur if an 8½" diameter impeller were used and the head (differential pressure) required by the process was low? Answer: Since the pressure differential required is low, the flow rate will increase and for the 8½" diameter impeller, the power required by the drier (BHP) will increase.

Therefore, a motor can trip out on overload, a steam turbine's speed can reduce or a diesel engine can trip on high engine temperature.

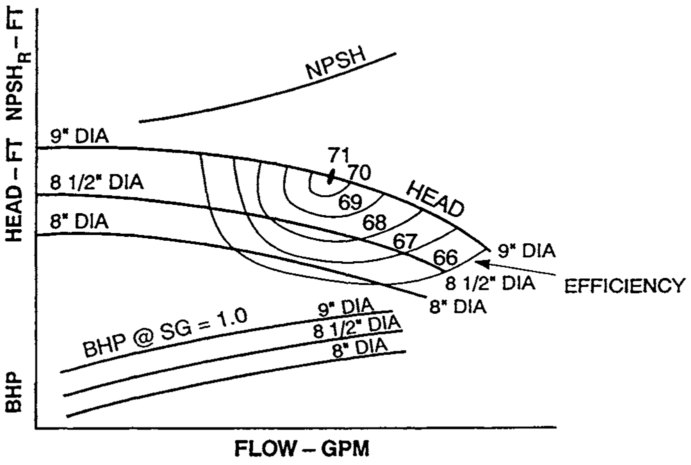


Figure 7.18 A typical centrifugal pump performance curve

These facts are shown in Figure 7.19.

Effect of the process on drivers

- Motors can trip on overload
- Steam turbines can reduce speed
- Diesel engines can trip on high engine temperature

Figure 7.19 Effect of the process on drivers

Auxiliary system reliability is also affected by process changes. Auxiliary systems support the equipment and their components by providing … clean, cold fluid to the components at the correct differential pressure, temperature and flow rate.

Typical auxiliary systems are:

- Lube oil systems
- Seal flush system
- Seal steam quench system
- Cooling water system

The reliability of machinery components (bearings, seals, etc.) is directly related to the reliability of the auxiliary system. In many cases, the root cause of the component failure is found in the supporting auxiliary system.

As an example, changes in auxiliary system supply temperature, resulting from cooling water temperature or ambient air temperature changes, can be the root cause of component failure. Figure 7.20 presents these facts.

Component (bearing and seal) reliability

- Is directly related to auxiliary system reliability
- Auxiliary system reliability is affected by process condition changes.
- 'Root causes' of component failure are often found in the auxiliary system.

Figure 7.20 Component (bearing and seal) reliability

As a result, the condition of all the auxiliary systems supporting a piece of equipment must be monitored. Please refer to Figure 7.21.

Always 'think system'

- Monitor auxiliary system condition
- Inspect auxiliary system during component replacement

Figure 7.21 Always 'think system'

The major machinery components

Please refer again to Figure 7.15 which shows how process condition changes can cause damage and/or failure to any pump component.

Regardless of the type of machinery, the major component classifications are the same. The major machinery components and their systems are shown in Figure 7.22.

Major machinery components and systems

- Rotor
- Radial bearing
- Thrust bearing
- Seal
- Auxiliary systems

Figure 7.22 Major machinery components and systems

Regardless of the type of machinery, if we monitor the condition of each major component and its associated system, we will know the condition of the machine!

Refer to Figure 7.23 and define the major components and their associated systems.

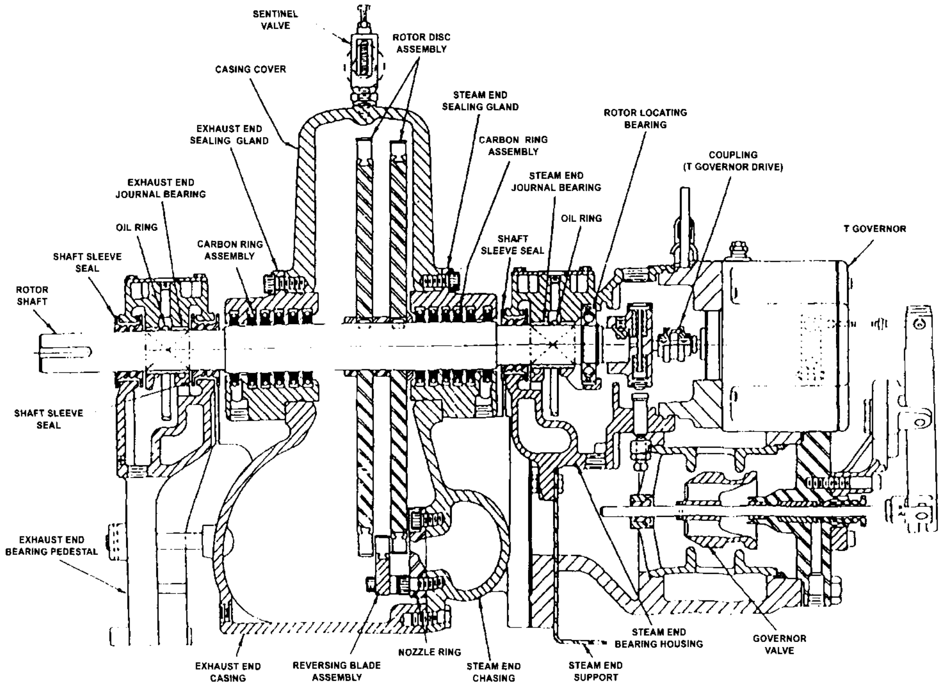

SENTINEL VALVE
ROTOR DISC ASSEMBLY
CASING COVER
STEAM END SEALING GLAND
ROTOR LOCATING BEARING
EXHAUST END SEALING GLAND
CARBON RING ASSEMBLY
COUPLING (T GOVERNOR DRIVE)
STEAM END JOURNAL BEARING
EXHAUST END JOURNAL BEARING
OIL RING
T GOVERNOR
OIL RING
CARBON RING ASSEMBLY
SHAFT SLEEVE SEAL
SHAFT SLEEVE SEAL
ROTOR SHAFT
SHAFT SLEEVE SEAL
EXHAUST END BEARING PEDESTAL
STEAM END BEARING HOUSING
GOVERNOR VALVE
NOZZLE RING
EXHAUST END CASING
REVERSING BLADE ASSEMBLY
STEAM END CHASING
STEAM END SUPPORT

Figure 7.23 Typical general purpose (pump drive) turbine assembly (Courtesy of Elliott Company)

The concept of fluid head

(Rotor component knowledge)

- Introduction
- Definition
- Paths of compression
- The different types of gas head
- Dynamic compressor curve format

Introduction

Without a doubt one of the most confused principles of turbocompressor design in my experience has been that of fluid head. I have found that understanding this concept can best be achieved by recognizing fluid head is the energy required to achieve specific process requirements.

Head is the energy in foot-pounds force required to compress and deliver one pound of a given fluid from one energy level to another. One of the confusing things about this concept is that the industry persists in defining head in feet. Head should be expressed in foot-pounds force per pound mass or British Thermal Units per pound. A British Thermal Unit per pound of fluid is equal to exactly 778 foot-pounds force per pound mass of that fluid.

Remember, when we deal with fluid head, a fluid can be either a liquid or a gas depending upon the conditions of the fluid at that time. Ethylene for instance, can be either a liquid or a gas depending on it's pressure and temperature. If it is a liquid, an ethylene pump will be used and the energy required to increase the pressure of the liquid from P_1 to P_2 will be defined as head in foot-pound force per pound mass. Conversely, if the conditions render it a vapor, an ethylene compressor would be used to achieve the same purpose. We will see in this section that the amount of energy required to compress a liquid or a gas the same amount will be significantly higher in favor of the gas because the gas is at a much lower density than that of a liquid. Understanding a Mollier Diagram is an important aid to understanding the concept of fluid head. Every fluid can have a Mollier Diagram drawing which expresses energy on the X axis and pressure on the Y axis for various temperatures. Increasing the pressure of a vapor will result in increased energy required.

Having defined the concept of head as that of energy, one must now investigate the different types of ideal (reversible) gas heads, namely isothermal, isentropic or adiabatic and polytropic. All of these types of fluid head simply describe the path that the gas takes in being compressed. It must be remembered that any type of head can be used to describe a reversible compressor path as long as the Vendor uses the appropriate head and efficiency in his data reduction calculations. In this section we will show the assumptions for various kinds of heads and the relative difference in their values. In addition, the definitions of each type will be stated.

Definition

The definition of head required by any fluid compression process is presented in Figure 7.24.

In any compression process, the amount of energy required to compress one pound of mass of a specific gas, at a given temperature from compressor suction flange (P_1) to the discharge flange (P_2) is defined as head required.

Figure 7.25 demonstrates how the density of a fluid significantly affects the amount of energy (head) required in a compression process.

Water with a density of 62.4 lbs/ft^3 requires only 231 ft-lb force per lb mass to compress the liquid 100 PSI. Note also, the equation for head

HEAD REQUIRED ***is the energy in foot pounds required to compress and deliver one pound of a given fluid from one energy level to another.***

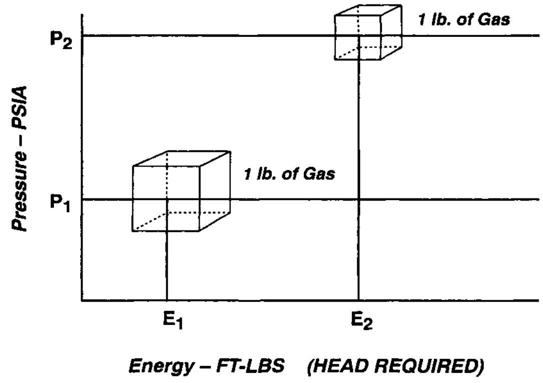


Figure 7.24 Head required definition

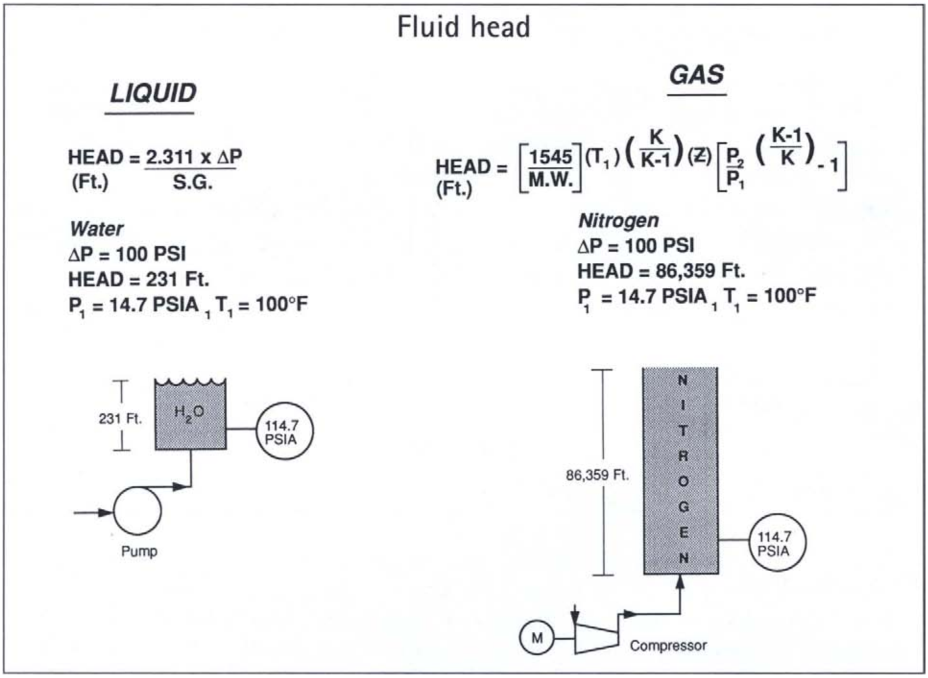

Fluid head
LIQUID
$\text{HEAD (Ft.)} = \frac{2.311 \times \Delta P}{\text{S.G.}}$
Water
ΔP = 100 PSI
HEAD = 231 Ft.
P_1 = 14.7 PSIA, T_1 = 100°F
231 Ft.
H_2O
114.7 PSIA
Pump
GAS
$\text{HEAD (Ft.)} = \left[\frac{1545}{\text{M.W.}}\right](T_1)\left(\frac{K}{K-1}\right)(Z)\left[\frac{P_2}{P_1}^{\left(\frac{K-1}{K}\right)} - 1\right]$
Nitrogen
ΔP = 100 PSI
HEAD = 86,359 Ft.
P_1 = 14.7 PSIA, T_1 = 100°F
86,359 Ft.
NITROGEN
114.7 PSIA
M
Compressor

Figure 7.25 Fluid head definition

required by a liquid is independent of temperature. On the other hand, nitrogen with a density of 0.07 lbs/ft^3 requires approximately 350 times the energy! This is because in both the case of water and nitrogen, the process requires that the fluid be compressed 100 PSI. However, since the mass of nitrogen is only 0.1% of the mass of water, a much greater amount of energy is required to compress the gas.

Figure 7.26 shows the head produced characteristics of positive displacement and dynamic compressors.

Note that regardless the amount of head required by the process, the flow rate of a positive displacement compressor is not affected. On the other hand, a dynamic compressor's flow rate is significantly affected by changes in the head required by the process. This is because the characteristic of any dynamic compressor is that it can only produce a greater amount of energy at a lower flow rate. The reason for this characteristic will be explained in a subsequent module. Therefore, any increase in the head required by a process will reduce the flow rate of a dynamic compressor. This is an extremely important fact because reduced flow rate in a dynamic compressor can lead to extreme, long term mechanical damage to the compressor unit. Also note in Figure 7.26 that the flatter the head produced curve a dynamic compressor possesses, the greater the effect of head required upon flow rate.

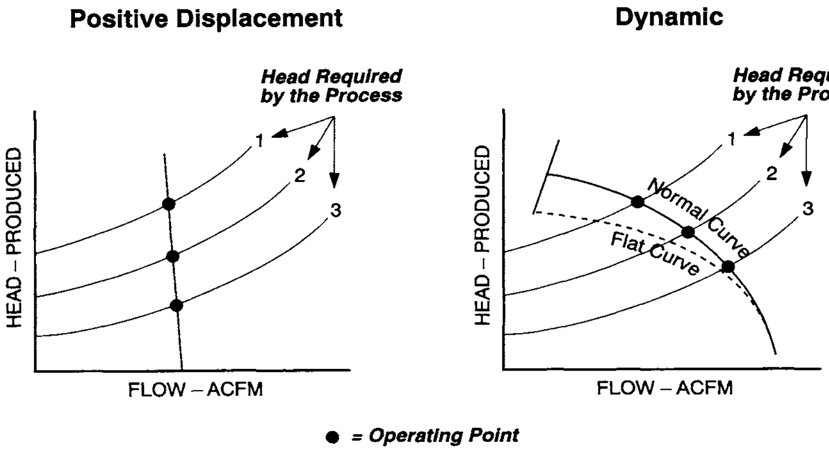


Figure 7.26 Compressor head produced characteristics

Head required

Thorough understanding of the concepts of head required by the process and head produced by the compressor is absolutely essential if dynamic compressor operation is to be understood.

It has been my experience that a lack of understanding exists in the area of dynamic compressor performance and often leads to a much greater emphasis upon the compressor's mechanical components (impellers, labyrinths, seals, bearings and shafts). In many cases, the root cause of dynamic compressor mechanical damage is that the head required by the process system exceeded the capability of the dynamic compressor.

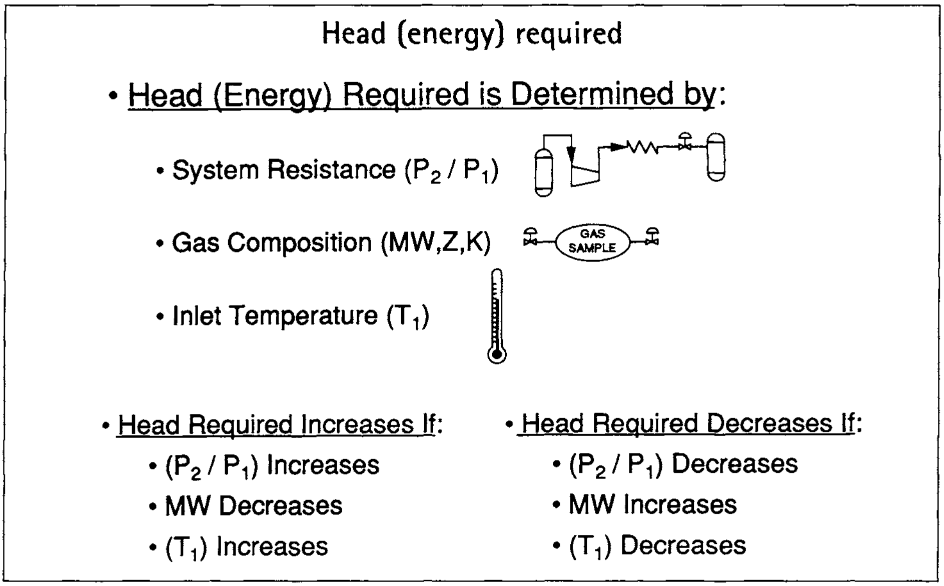
Figure 7.27 presents the factors that determine the head (energy) required by any process.

Note that the head required by the process is inversely proportional to the gas density. If the gas density decreases, the head required by the process will increase. Gas density will decrease if gas temperature increases, inlet pressure decreases or molecular weight decreases. If the head required by the process increases, the flow rate of any dynamic compressor will decrease as shown in Figure 7.26. If the gas density increases, the head required by the process will decrease and dynamic compressor flow rate will increase.

Head produced

In Figure 7.28, the factors that determine the head produced by a dynamic compressor are presented.

Simply stated, for a given impeller vane shape, head produced by a dynamic compressor is a function of impeller diameter and impeller



Head (energy) required

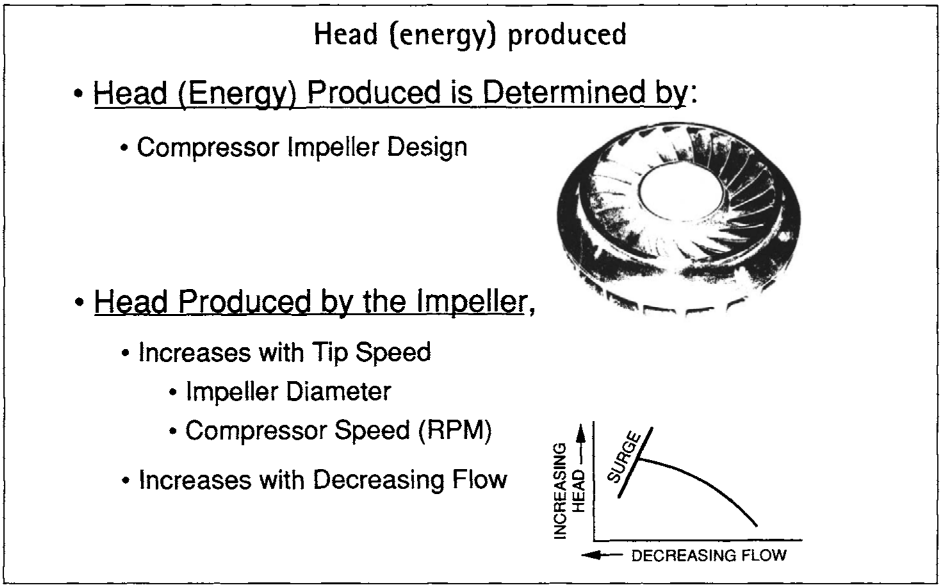
- Head (Energy) Required is Determined by:
  - System Resistance (P_2 / P_1)
  - Gas Composition (MW,Z,K)
  - Inlet Temperature (T_1)

- Head Required Increases If:
  - (P_2 / P_1) Increases
  - MW Decreases
  - (T_1) Increases
- Head Required Decreases If:
  - (P_2 / P_1) Decreases
  - MW Increases
  - (T_1) Decreases

Figure 7.27 Head (energy) required

speed. Once the impeller is designed, it will produce only one value of head for a given shaft speed and flow rate.

The only factor that will cause a lower value of head to be produced than stated by the compressor performance curve is if the compressor has experienced mechanical damage or if it is fouled.



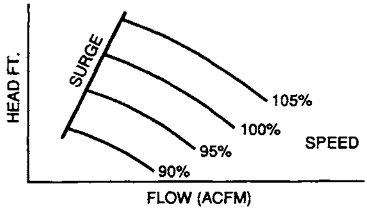
Head (energy) produced

- Head (Energy) Produced is Determined by:
  - Compressor Impeller Design
- Head Produced by the Impeller,
  - Increases with Tip Speed
    - Impeller Diameter
    - Compressor Speed (RPM)
  - Increases with Decreasing Flow

Figure 7.28 Head (energy) produced

The turbo-compressor curve

- The Turbo-Compressor Head (Energy) vs. Flow Curve is *Fixed by Compressor Design*



- For a Given Speed and Flow, If an Operating (Test) Point *Does Not Fall on the Curve*,* the Unit Requires Maintenance

* *Within ± 10% (Accounting for Instrument Calibration Error)*

Figure 7.29 The turbo-compressor curve

Figure 7.29 shows how the need for dynamic compressor inspection can be determined.

If for a given flow rate and shaft speed, the head produced falls below the value predicted by greater than 10%, the compressor should be inspected at the first opportunity. Having explained the concept of head required by the process, the method of calculating the head required by a process system needs to be discussed.

Paths of compression

Figure 7.30 presents a typical Mollier Diagram plotted pressure vs energy.

A Mollier Diagram can be drawn for any pure fluid or fluid mixture. Usually, Mollier Diagrams are prepared only for pure fluids since any change in fluid mixture will require a new Mollier Diagram to be prepared. The Mollier Diagram can be used to determine the head required by the process system for any liquid, saturated vapor or vapor compression process. Observe that for liquid compression, the amount of energy required to increase the pressure from P_1 to P_2 is very small. However, for compression of a vapor, the amount of energy required to compress from P_1 to P_2 is very large as previously explained. Refer back to Figure 7.26 of this module and study the equations that are used to

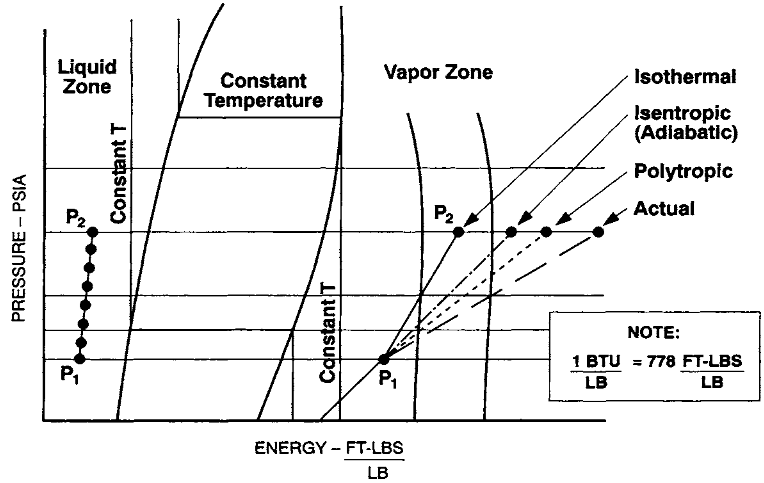

Liquid Zone
Constant T
Constant Temperature
Vapor Zone
Constant T
PRESSURE – PSIA
P2
P1
P2
P1
Isothermal
Isentropic (Adiabatic)
Polytropic
Actual
NOTE:
1 BTU / LB = 778 FT-LBS / LB
ENERGY – FT-LBS / LB

Figure 7.30 Ideal paths of compression

determine the head required for a liquid and a vapor. In addition to many more parameters being required for calculation of head required for a vapor, certain ideal assumptions must be made.

The different types of gas head

A vapor can be ideally compressed by any one of the following reversible thermodynamic paths:

- Isothermal – constant temperature
- Isentropic (adiabatic) – no heat loss
- Polytropic – temperature not constant and heat lost

The actual path that any compressor follows in compressing a vapor from P_1 to P_2 is equal to the reversible path divided by the compressor's corresponding path efficiency. Therefore, we can write the following equation:

$$\textit{Actual Head} = \frac{\textit{Head Isothermal}}{\textit{Eff'y. Isothermal}} = \frac{\textit{Head Isentroic}}{\textit{Eff'y. Isentropic}} = \frac{\textit{Head Polytropic}}{\textit{Eff'y. Polytropic}}$$

When evaluating compressor bids from different vendors quoting different types of reversible heads the above equation proves useful. If the head produced by one vendor divided by the corresponding

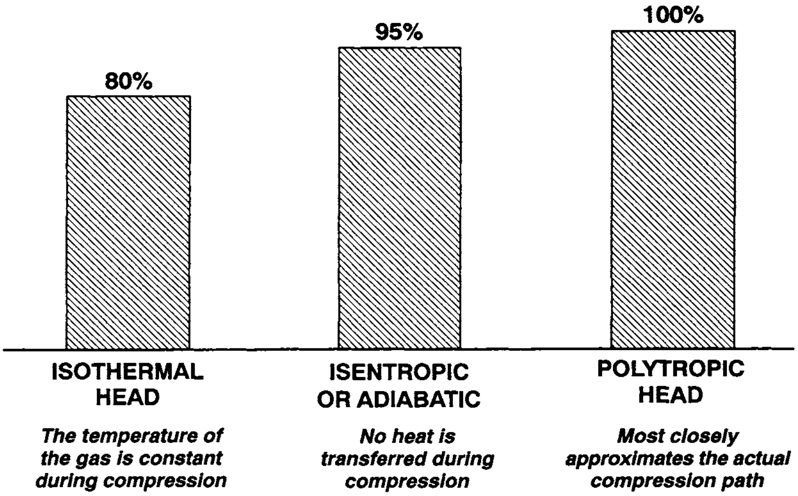


Figure 7.31 Ideal compression heads – relative values

efficiency is not equal to the corresponding values quoted by the competition . . . Better start asking why!

Figure 7.31 defines the different types of ideal heads and shows the relative difference between their values as compared to polytropic head.

The definition of polytropic head is confusing and difficult to understand if not investigated further.

The ideal gas head equations are described in Figure 7.32.

Note that the only difference between isentropic and polytropic head is the values;

$$\frac{K-1}{K} \quad \text{and} \quad \frac{n-1}{n}$$

Also note that $\frac{n-1}{n} = \frac{\frac{K-1}{K}}{\eta \text{ poly}}$

Now if η poly = 100%, $\frac{n-1}{n} = \frac{K-1}{K}$ or

Polytropic Head = Isentropic Head

Therefore, I think of $\frac{n-1}{n}$ as a correction

Factor to $\frac{K-1}{K}$ that will most closely approximate the actual

compressor path for a given compressor (refer back to Figure 7.30)

Ideal gas head equations

Isothermal

$$HD = \left(\frac{1546}{M.W}\right)(T_1)(Z_{AVG})\left[L_N\left(\frac{P_2}{P_1}\right)\right]$$

Isentropic (Adiabatic)

$$HD = \left(\frac{1545}{M.W}\right)(T_1)\left(\frac{K}{K-1}\right)(Z_{AVG})\left[\left(\frac{P_2}{P_1}\right)^{\frac{(K-1)}{K}}-1\right]$$

Polytropic

$$HD = \left(\frac{1545}{M.W}\right)(T_1)\left(\frac{n}{n-1}\right)(Z_{AVG})\left[\left(\frac{P_2}{p1}\right)^{\frac{(n-1)}{n}}-1\right]$$

Where:

$\frac{1545}{M.W}$ = Gas constant "R"

T_1 = Inlet temp. - °R

°R = 460 + °F

Z_{AVG} = Average compressibility $\frac{(Z_1 + Z_2)}{2}$

Ln = Log to base e

P_2 = Discharge Pressure - PSIA

P_1 = Inlet pressure - PSIA

K = Ratio of specific heats C_p/C_v

n = Polytropic exponent

$\frac{(n-1)}{n} = \left(\frac{K-1}{K}\right)\left(\frac{1}{\eta_{POLY}}\right)$

η_{POLY} = Polytropic efficiency

Figure 7.32 Ideal gas head equations

Remember, polytropic head is an ideal reversible compression path. Today, most compressor vendors have adopted polytropic head as their standard for multistage compressors. The main reason is that polytropic head, since it contains an efficiency term, allows individual impeller (stage) heads to be added.

Dynamic compressor curves format

Finally, Figure 7.33 presents the different ways that compressor compression performance can be formatted.

Head vs flow is always preferred because the head produced by a dynamic compressor is not significantly affected by gas density. However, compression ratio or discharge pressure are! That is, compression ratio and discharge pressure curves are invalid if the inlet gas temperature, inlet pressure or molecular weight changes! This may seem confusing at first, but refer back to Figure 7.28 which shows how head is produced by a dynamic compressor. It is a function only of impeller diameter and shaft speed. Gas density influences the head required by the process (refer to Figure 7.27).

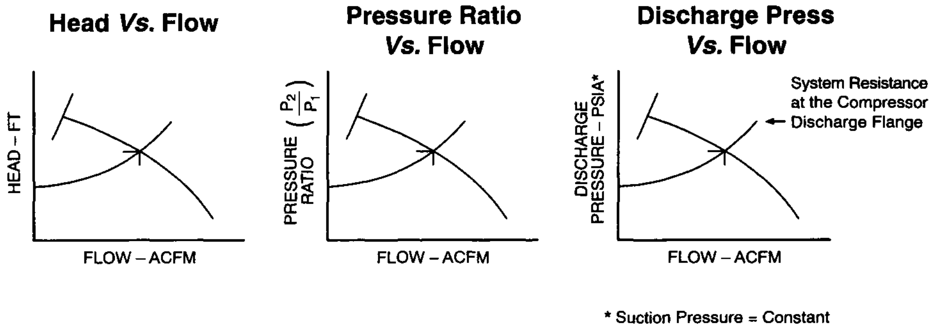


Figure 7.33 Dynamic compressor curves format

Radial bearing design

(Journal bearing component knowledge)

- Introduction
- Anti-friction bearings
- Hydrodynamic bearings
- Hydrodynamic bearing types
- Condition monitoring
- Vibration instabilities

Introduction

Radial bearings fall into two major categories:

- Anti-friction
- Hydrodynamic

Anti-friction bearings rely on rolling elements to carry the load of the equipment and reduce the power losses or friction.

Hydrodynamic bearings rely on a liquid film, usually lubricating oil, to carry the load of the equipment and minimize friction. In general, anti-friction bearings are used for equipment of low horsepower. Hydrodynamic bearings are generally used for all rotating equipment above approximately 500 horsepower. During this section we will concentrate on the subject of hydrodynamic bearings since they are the principle bearing type used in turbo-compressor operation.

A bearing in the radial position is responsible for mainly supporting the weight of the shaft and any dynamic loads that are present in the system. It can be generally stated that the dynamic loads are in the order of 20% of the static loads on the bearing journal. In the case of gears however, the radial load component is made up principally of the meshing force of the gear teeth and the load will vary from zero to maximum torque. One must be careful in this design to assure that the various load angles occur in zones of the bearing that can support these loads.

We will examine loads on hydrodynamic bearings and present an example of the bearing sizing based on static and dynamic loads of a rotor system. We will see that there are specific oil film pressure limits which dictate the bearing dimensions (length and diameter).

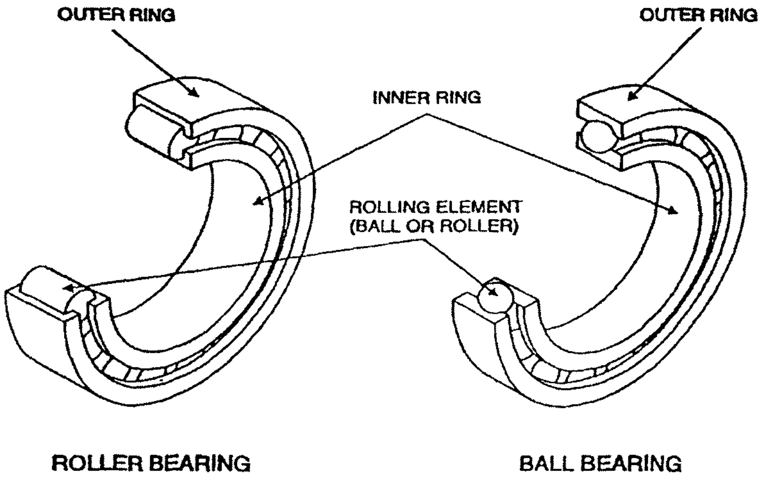


Figure 7.34 Anti-friction bearings

The types of hydrodynamic bearings will be reviewed; plain, stationary anti-whirl types and tilt pads.

We will conclude this section by discussing the condition monitoring requirements for radial bearings and briefly discuss shaft vibration and vibration troubleshooting (diagnostics).

Anti-friction bearings

Anti-friction radial bearings support the rotor using rolling elements to reduce friction losses. They are used in low horsepower applications (below 500 H.P.) and in aero-derivative gas turbines. Examples of roller and ball type anti-friction bearings are shown in Figure 7.34.

As previously mentioned, all bearings are designed on the basis of sufficient bearing area to support all the forces acting on the bearing. That is:

$$P = \frac{F}{A}$$

Where: P = Pressure on the bearing elements – P.S.I.
F = The total of all static and dynamic forces acting on the bearing
A = Contact area

For anti-friction bearing applications, the pressure, P is the point contact or ‘Hertzian’ stress on the bearing elements and rings or ‘races’. For an anti-friction bearing to be properly designed, its D-N number and bearing life must be determined. Figure 7.35 presents the definition of D-N number and its uses.

D-N Number

- Is a measure of the rotational speed of the anti-friction bearing elements
- D-N number = bearing bore (millimeters) x speed (RPM)
- Approximate lubrication ranges

Lubrication type	**D-N Range**
Sealed	below 100,000
Regreaseable	100,000-300,000
Oil lube (unpressurized)	300,000-500,000
Oil lube (pressurized)	above 500,000

Figure 7.35 D-N number

Each type of anti-friction bearing has a maximum operating D-N number. If this value is exceeded, rapid bearing failure can occur. In addition, D-N numbers are typically used to determine the type of lubrication required for bearings. A common practice in the turbomachinery industry has been to use hydrodynamic bearings when the D-N number exceeds approximately 500,000.

The exception to this rule is aircraft gas turbines which can have D-N numbers in excess of 2,000,000. In these applications, hydrodynamic bearings are not used since the size and weight of the required lubrication system would be prohibitive.

Anti-friction bearings possess a finite life which is usually specified as 'B-10' or 'L-10' life as defined in Figure 7.36.

'B' OR 'L' – 10 Life

'B' or 'L' – 10 Life is defined as the life in hours that 9 out of 10 randomly selected bearings would exceed in a specific application.

'B' or 'L' – 10 Life =

$$\frac{16700}{N}\left[\frac{C}{F}\right]^3$$

Where: N = RPM
C = Load in LBS that will result in a bearing element life of 1,000,000 revolutions.
F= Actual load in LBS

Figure 7.36 'B' or 'L' – 10 life

An important fact to note is that the life of any anti-friction bearing is inversely proportional to cube of the bearing loads. As a result, a small change in the journal bearing loads can significantly reduce the bearing life. When anti-friction bearings suddenly start failing where they did not in the past, investigate all possible sources of increased bearing loads (piping forces, foundation forces, misalignment, unbalance, etc.). Anti-friction bearings are usually designed for a minimum life of 25,000 hours continuous operation.

Hydrodynamic bearings

Hydrodynamic bearings support the rotor using a liquid wedge formed by the motion of the shaft (see Figure 7.37).

Oil enters the bearing at supply pressure values of typically 15–20 psig. The shaft acts like a pump which increases the support pressure to form a wedge. The pressure of the support liquid (usually mineral oil) is determined by the area of the bearing by the relationship:

$$P = \frac{F}{A}$$

Where: P = Wedge support pressure (P.S.I.)
F = Total bearing loads (static and dynamic)
A = Projected bearing area ($A_{\mathrm{PROJECTED}}$)

$$A_{\mathrm{PROJECTED}} = L \times d$$

Where: L = Bearing axial length
d = Bearing diameter

As an example, a 4" diameter bearing with an axial length of 2" ($L/d = .5$) would have $A_{\mathrm{PROJECTED}} = 8\ \mathrm{in}^2$.

If the total of static and dynamic forces acting on the bearing are 1600 lbs. force, the pressure of the support wedge is:

$$P = \frac{1600\ \mathrm{Lb}_{\mathrm{FORCE}}}{8\ \mathrm{in}^2}$$
$$= 200\ \mathrm{P.S.I.}$$

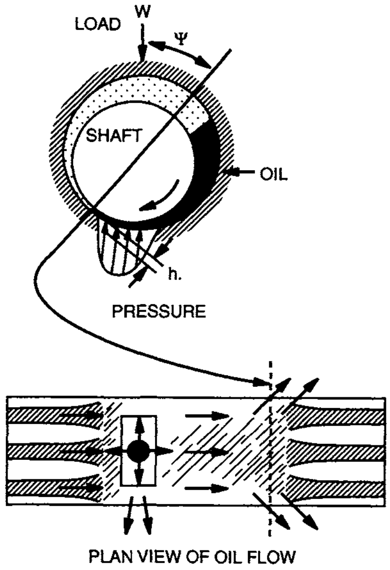


Figure 7.37 Hydrodynamic Lubrication (Courtesy of Bently Nevada Corp.)

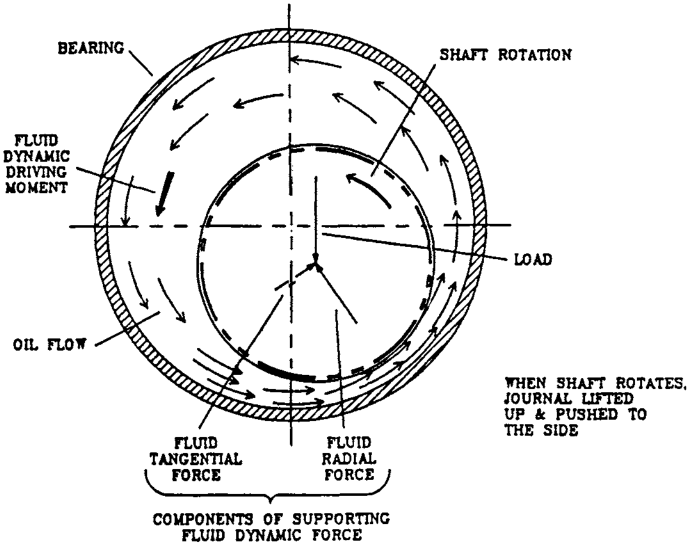

BEARING
SHAFT ROTATION
FLUID DYNAMIC DRIVING MOMENT
LOAD
OIL FLOW
WHEN SHAFT ROTATES, JOURNAL LIFTED UP & PUSHED TO THE SIDE
FLUID TANGENTIAL FORCE
FLUID RADIAL FORCE
COMPONENTS OF SUPPORTING FLUID DYNAMIC FORCE

Figure 7.38 Shaft/bearing dynamics (Courtesy of Bently Nevada Corp.)

The maximum desired design wedge pressure for oil is approximately 500 psi. However, it has been common practice to limit hydrodynamic bearing loads to approximately 250 psi in compressor applications. Figure 7.38 is a side view of a simple hydrodynamic bearing showing the dynamic load forces.

The primary force is the load which acts in the vertical direction for horizontal bearings. However, the fluid tangential force can become large at high shaft speeds. The bearing load vector then is the resultant of the load force and fluid tangential force. The fluid radial force opposes the load vector and thus supports the shaft. It has been demonstrated that the average velocity of the oil flow is approximately 47–52% of the shaft velocity. The fluid tangential force is proportional to the journal oil flow velocity. If the fluid tangential force exceeds the load force, the shaft will become unstable and will be moved around the bearing shell. This phenomena is known as oil whirl.

Hydrodynamic bearing types

Regardless of the type of hydrodynamic bearing, all bearing surfaces are lined with a soft, surface material made of a composition of tin and lead. This material is known as Babbitt. Its melting temperature is above 400°F, but under load will begin to deform at approximately 320°F. Typical thickness of Babbitt over steel is 0.060 (1.5mm). Bearing embedded temperature probes are a most effective means of

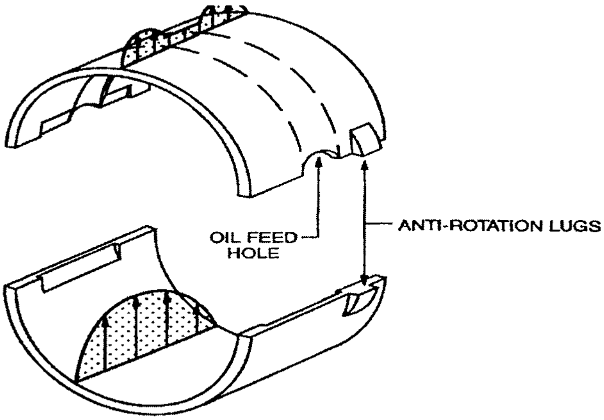


Figure 7.39 Straight sleeve bearing liner (Courtesy of Elliott Co.)

measuring bearing load point temperature and are inserted just below the Babbitt surface. RTD's or thermocouples can be used. There are many modifications available to increase the load effectiveness of hydrodynamic bearings. Among the methods available are:

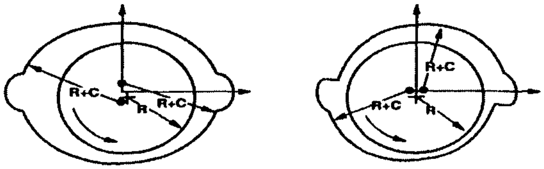
- Copper backed Babbitt or 'Trimetal' – to aid in heat removal
- Back pad cooling – used on tilt pad bearings to remove heat
- Direct cooling – directing cool oil to maximum load points

A typical straight sleeve hydrodynamic journal bearing is shown in Figure 7.39.

Straight sleeve bearings are used for low shaft speeds (less than 5,000 RPM) or for older turbo-compressor designs. Frequently, they are modified to incorporate a pressure dam, in the direction of rotation. The pressure dam must be positioned in the top half of the bearing to increase the load vector (see Figure 7.38). This action assures that the tangential force vector will be small relative to the load vector thus preventing shaft instability. It should be noted that incorrectly assembling the pressure dam in the lower half of the bearing would render this type of bearing unstable. When shaft speed is high, other alternatives to prevent rotor instabilities are noted in Figure 7.40.

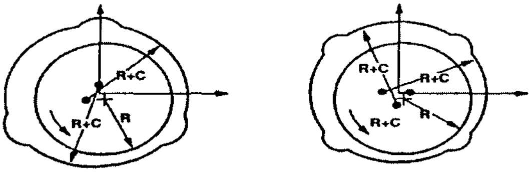
Shown are examples of anti-whirl bearings. The most common types of these bearings are the 3 and 4 lobe design. Elliptical and offset bearing designs do prevent instabilities but tend to increase shaft vibration if the load vector passes through the major axis of the bearing. These types of bearings may have to be rotated in the bearing brackets to prevent this occupance.

The most common hydrodynamic bearing for higher speed applications is the tilt pad journal bearing shown in Figure 7.41.



a) Elliptical or lemon bore bearing

b) Offset half bearing



c) Three lobe bearing

d) Four lobe bearing

Figure 7.40 Prevention of rotor instabilities

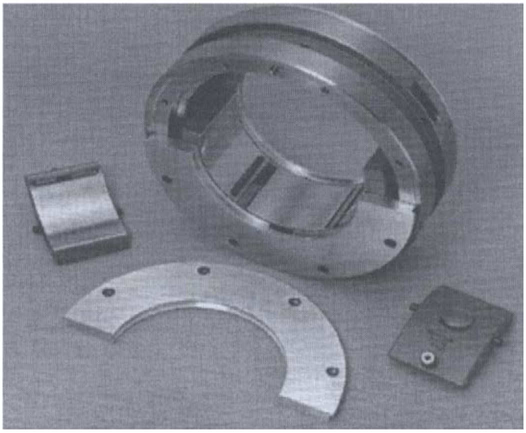


Figure 7.41 Tilting pad journal bearing assembly (Courtesy of Kingsbury, Inc.)

A tilting pad bearing offers the advantage of increased contact area since the individual pads conform to the shaft orbit. In addition, this type is also a highly effective anti-whirl bearing since the spaces between the pads prevent oil whirl.

Most end users specify tilt pad radial and thrust bearings for turbo-compressor applications.

Figure 7.42 shows the mechanical frictional losses and oil flow requirements for a tilt pad journal bearing as a function of shaft speed.

Note that the basis for horsepower loss and oil flow is an oil temperature rise of 30°F. This is the normal design ΔT for all

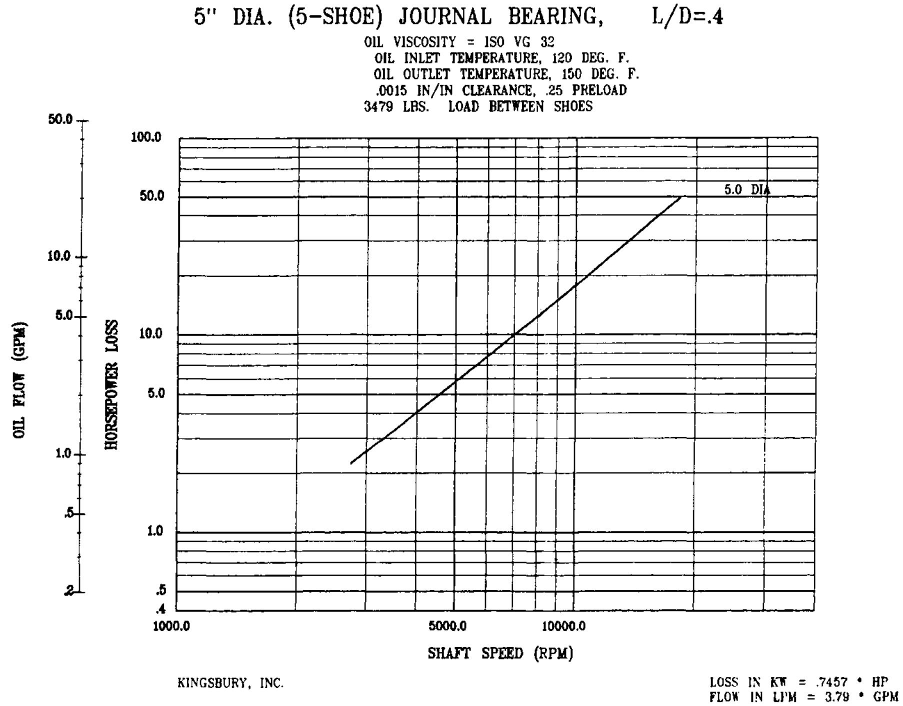


Figure 7.42 Typical journal bearing selection curve (Courtesy of Kingsbury, Inc.)

hydrodynamic bearings. Also given in this figure is the data necessary to calculate bearing pressure at the load point.

As an exercise calculate the following for this bearing:

- Projected Area

$A_{PROJECTED}$ = 5" x 2"
= 10 square inches

- Pressure

= 3479 Lb force = 10 square inches
= 347.9 psi on the oil film at load point

Condition monitoring

In order to determine the condition of any journal bearing, all the parameters that determine its condition must be monitored.

Figure 7.43 presents the eight parameters that determine the condition of a hydrodynamic journal bearing along with typical limits. Attendees are advised to consult the manufacturers instruction book for vendor recommended limits.

Parameter		Limits
1	Radial vibration (peak to peak)	2.5 mils (60 microns)
2	Bearing pad temperature	220°F (108°C)
3	Radial shaft position (except for gearboxes where greater values are normal from unloaded to loaded operation)	>30° change and/or 30% position change
4	Lube oil supply temperature	140°F (60°C)
5	Lube oil drain temperature	190°F (90°C)
6	Lube oil viscosity	Off spec 50%
7	Lube oil flash point	Below 200°F (100°C)
8	Lube oil particle size	Greater than 25 microns

Condition monitoring parameters and their alarm limits according to component:

1. Journal bearing (hydrodynamic)

Figure 7.43 The eight parameters that determine the condition of a hydrodynamic journal bearing along with typical limits

One important parameter noted in Figure 7.43 that is frequently overlooked is shaft position. Change of shaft position can only occur if the forces acting on a bearing change or if the bearing surface wears. Figure 7.44 shows how shaft position is determined using standard shaft proximity probes.

Regardless of the parameters that are condition monitored, relative change of condition determines if and when action is required. Therefore, effective condition monitoring requires the following action for each monitored condition.

- Establish baseline condition
- Record condition trend
- Establish condition limit

Figure 7.45 presents these facts for a typical hydrodynamic journal bearing.

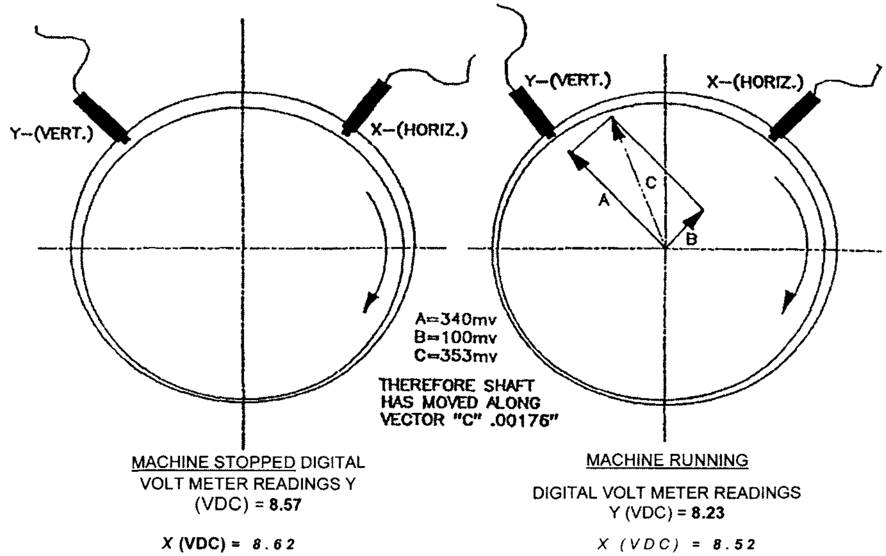

Y–(VERT.)
X–(HORIZ.)
Y–(VERT.)
X–(HORIZ.)
A
B
C
A=340mv
B=100mv
C=353mv
THEREFORE SHAFT HAS MOVED ALONG VECTOR "C" .00176"
MACHINE STOPPED DIGITAL VOLT METER READINGS Y (VDC) = 8.57
X (VDC) = 8.62
MACHINE RUNNING
DIGITAL VOLT METER READINGS Y (VDC) = 8.23
X (VDC) = 8.52

Figure 7.44 Shaft movement analysis (relative to bearing bore) (Courtesy of M.E. Crane Consultant)

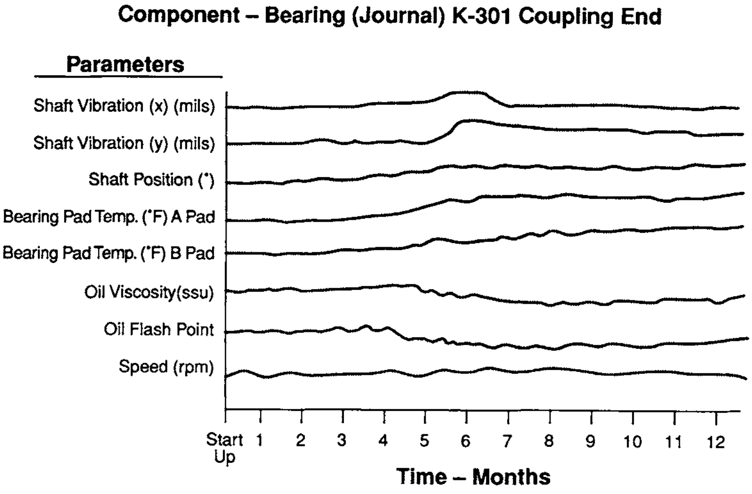


Figure 7.45 Trending data for a typical hydrodynamic journal bearing

Based on the information shown in this trend, the bearing should be inspected at the next scheduled shutdown. A change in parameters during month 6 has resulted in increased shaft position, vibration and bearing pad temperature.

Vibration instabilities

Vibration is an important condition associated with journal bearings because it can provide a wealth of diagnostic information valuable in determining the root cause of a problem.

Figure 7.46 presents important information concerning vibration.

Vibration

- Vibration is the result of a system being acted on by an excitation.
- This excitation produces a dynamic force by the relationship:

$$F_{DYNAMIC} = Ma$$

Where: M = Mass (Weight/g)
g = Acceleration due to gravity (386 IN/SEC^2)
a = Acceleration of mass M (IN/SEC^2)

- Vibration can be:
  - Lateral ↕
  - Axial → ←
  - Torsional

Figure 7.46 Vibration

Figure 7.47 defines excitation forces with examples that can cause rotor (shaft) vibration.

Turbo-compressors generally monitor shaft vibration relative to the bearing bracket using a non-contact or ‘proximity probe’ system as shown in Figure 7.48. The probe generates a D.C. eddy current which continuously measures the change in gap between the probe tip and the shaft. The result is that the peak to peak unfiltered (overall) shaft vibration is read in mils or thousandth of an inch. The D.C. signal is normally calibrated for 200 milli volts per mil. Probe gaps (distance between probe and shaft) are typically 0.040 mils or 8 volts D.C. to assure the calibration curve is in the linear range. It is important to remember that this system measures shaft vibration relative to the bearing bracket and assumes the bearing bracket is fixed. Some systems incorporate an additional bearing bracket vibration monitor and thus record vibration relative to the earth or ‘seismic vibration’.

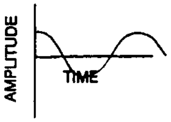
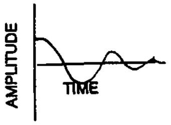
CATEGORY	EXAMPLES	EXCITATION TYPE	VIBRATION TYPE
FORCED VIBRATIONS 	UNBALANCE MISALIGNMENT PULSATION	CONSTANT CONSTANT PERIODIC	LATERAL LATERAL AND AXIAL TORSIONAL
TRANSIENT VIBRATIONS 	SYNCHRONOUS MOTOR START-UP IMPULSE (SHOCK FORCE)	RANDOM RANDOM	TORSIONAL TORSIONAL, AXIAL, RADIAL
SELF EXCITED	INTERNAL RUB OIL WHIRL GAS WHIRL	RANDOM CONSTANT CONSTANT	LATERAL LATERAL LATERAL

Figure 7.47 Excitation forces with examples

As previously discussed, vibration limits are usually defined by:

$$\text{Vibration(mils p-p)} = \sqrt{\frac{12000}{RPM}}$$

This value represents the allowable shop acceptance level. A.P.I. recommends alarm and trip shaft vibration levels be set as follows:

$$V_{ALARM} = \sqrt{\frac{24000}{RPM}}$$

$$V_{TRIP} = \sqrt{\frac{36000}{RPM}}$$

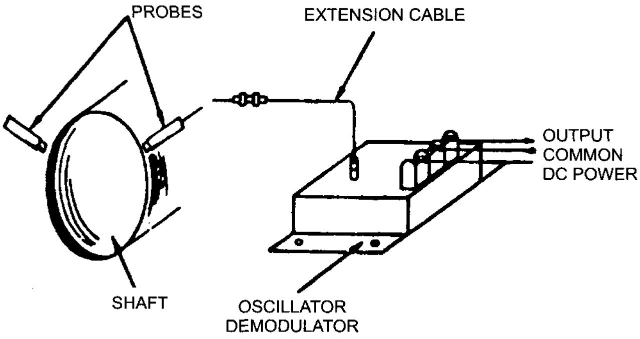


Figure 7.48 Non contact displacement measuring system

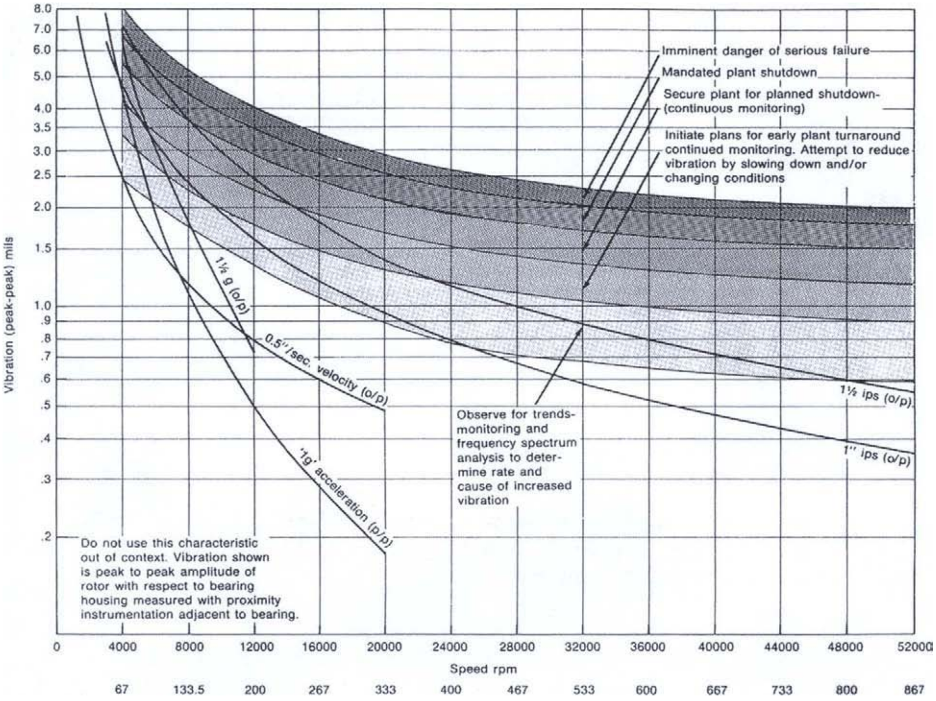


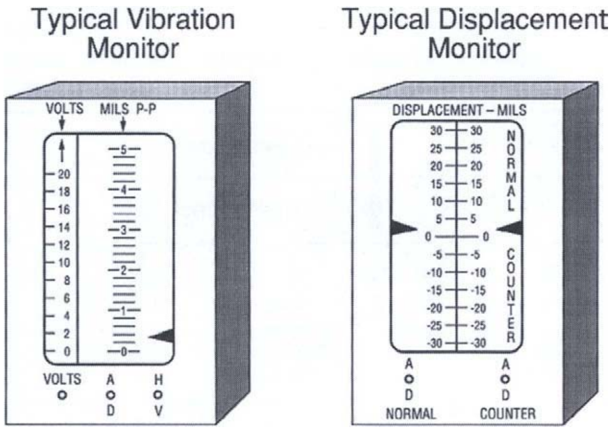
Figure 7.49 Vibration severity chart (Courtesy of Dresser-Rand and C. J. Jackson P.E.)

In the writers' opinion, shaft vibration alarm and trip levels should be based on the following parameters as a minimum and should be discussed with the machinery vendor prior to establishing levels:

- Application (critical or general purpose)
- Potential loss of revenue
- Application characteristics (prone to fouling, liquid, unbalance, etc.)
- Bearing clearance
- Speed
- Rotor actual response (Bode Plot)
- Rotor mode shapes (at critical and operating speeds)

Figure 7.49 presents a vibration severity chart with recommended action.

A schematic of a shaft vibration and shaft displacement monitor are shown in Figure 7.50.

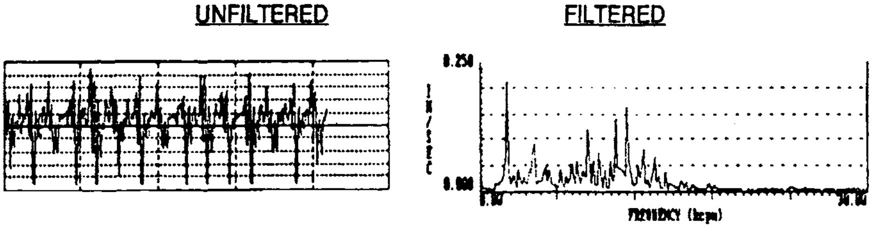

Typical Vibration Monitor

Typical Displacement Monitor

VOLTS
MILS P-P
20
18
16
14
12
10
8
6
4
2
0
5
4
3
2
1
0
VOLTS
A
O
D
H
O
V
DISPLACEMENT – MILS
30
25
20
15
10
5
0
-5
-10
-15
-20
-25
-30
NORMAL
COUNTER
A
O
D
NORMAL
A
O
D
COUNTER

Figure 7.50 Shaft vibration and displacement

ANY VIBRATION SIGNAL IS MADE UP OF ONE OR MORE FREQUENCIES. A TYPICAL UNFILTERED (OVERALL) AND FILTERED SIGNAL ARE SHOWN BELOW:


UNFILTERED

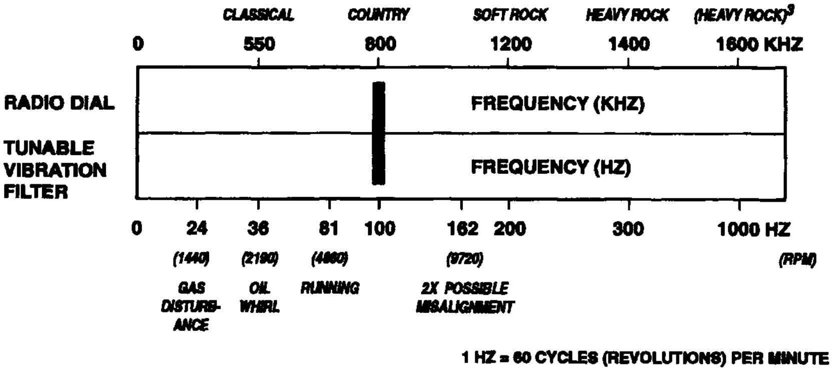
FILTERED

AN ANALOGY TO A FILTERED SIGNAL IS A RADIO. IN A GIVEN LOCALITY, MANY STATIONS ARE TRANSMITTING SIMULTANEOUSLY. ANY GIVEN STATION IS OBTAINED BY ADJUSTING THE TUNER TO THE CORRECT TRANSMISSION FREQUENCY.

Figure 7.51 Vibration frequency

As mentioned above, vibration is measured unfiltered or presents ‘overall vibration’. Figure 7.51 shows a vibration signal in the unfiltered and filtered conditions. All vibration diagnostic work (troubleshooting) relies heavily on filtered vibration to supply valuable information to determine the root cause of the vibration.

Figure 7.52 presents an example of a radio tuner as an analogy to a filtered vibration signal.


CLASSICAL
COUNTRY
SOFT ROCK
HEAVY ROCK
(HEAVY ROCK)?
0
550
800
1200
1400
1600 KHZ
RADIO DIAL
FREQUENCY (KHZ)
TUNABLE VIBRATION FILTER
FREQUENCY (HZ)
0
24
36
81
100
162
200
300
1000 HZ
(1440)
(2160)
(4860)
(9720)
(RPM)
GAS DISTURB-ANCE
OIL WHIRL
RUNNING
2X POSSIBLE MISALIGNMENT
1 HZ = 60 CYCLES (REVOLUTIONS) PER MINUTE

By Adjusting the Tuner (Filter) to a Selected Station (Frequency) the Desired Program (Vibration Frequency Signal) can be Obtained if it is "On the Air" (Present)

Figure 7.52 Radio tuner/vibration filter analogy

By observing the predominant filtered frequencies in any overall (unfiltered) vibration signal, valuable information can be gained to add in the troubleshooting procedure and thus define the root cause of the problem.

Rotor axial (thrust) forces

(Thrust bearing component knowledge)

- Introduction
- The hydrodynamic thrust bearing
- Impeller thrust forces
- Rotor thrust balance
- Thrust condition monitoring

Introduction

In every rotating machine utilizing reaction type blading, a significant thrust is developed across the rotor by the action of the impellers or blades. Also in the case of equipment incorporating higher than atmospheric suction pressure, a thrust force is exerted in the axial direction as a result of the pressure differential between the pressure in the case and atmospheric pressure.

In this section we will cover a specific rotor thrust example and calculate thrust balance for a specific case. We will see the necessity in some applications of employing an axial force balance device known as a balance drum. In many instances, the absence of this device will result in excessive axial (thrust) bearing loadings. For the case of a machine with a balance device, the maintenance of the clearances on this device are of utmost importance. In many older designs the clearances are maintained by a fixed close clearance bushing made out of babbitt which has a melting temperature of approximately 350°F, depending on the pressure differential across the balance drum. If the temperature in this region should exceed this value, the effectiveness of the balance drum would suddenly be lost and catastrophic failures can occur inside the machine. Understanding the function of this device and the potential high axial forces involved in its absence is a very important aspect of condition monitoring of turbo-compressors.

We will also examine various machine configurations including natural balanced (opposed) thrust and see how thrust values change even in the case of a balanced machine as a function of machine flow rate.

Finally, we will examine thrust system condition monitoring and discuss some of the confusion that results with monitoring these machines.

The hydrodynamic thrust bearing

A typical hydrodynamic double acting thrust bearing is pictured in Figure 7.53.

The thrust bearing assembly consists of a thrust collar mounted on the rotor and two sets of thrust pads (usually identical in capacity) supported by a base ring (Michell Type).

The Kingsbury type includes a set of leveling plates between each set of pads and the base ring. This design is shown in Figure 7.54.

Both the Michell and Kingsbury types are used. Figure 7.55 provides a view of the leveling plates providing the self-equalizing feature in the Kingsbury design. The self-equalizing feature allows the thrust pads to lie in a plane parallel to the thrust collar.

Regardless of the design features, the functions of all thrust bearings are:

- To continuously support all axial loads
- To maintain the axial position of the rotor

The first function is accomplished by designing the thrust bearing to provide sufficient thrust area to absorb all thrust loads without exceeding the support film (oil) pressure limit (approximately 500 psi).

Figure 7.56 shows what occurs when the support film pressure limit is exceeded.

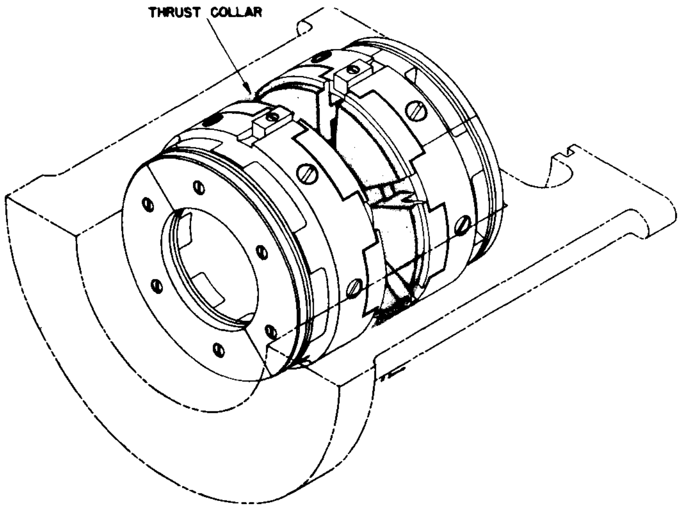


Figure 7.53 Double acting self-equalizing thrust bearing assembly (thrust collar removed) (Courtesy of Elliott Company)

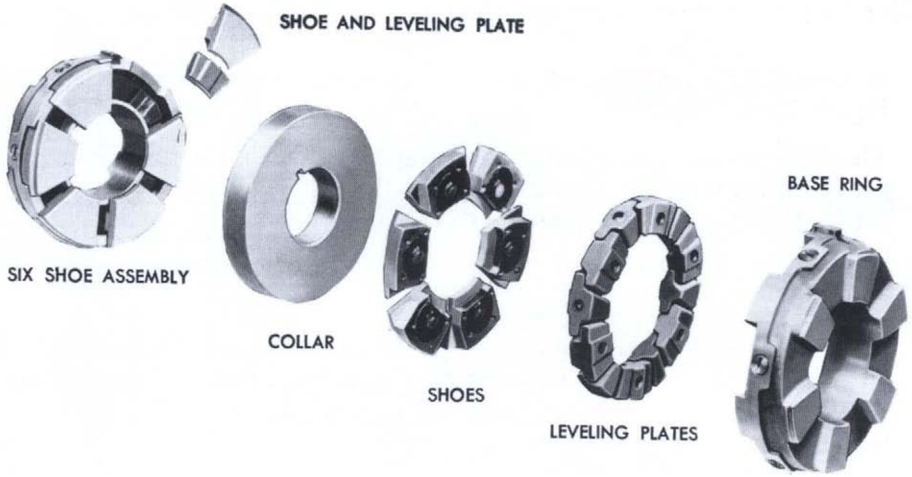

SHOE AND LEVELING PLATE
SIX SHOE ASSEMBLY
COLLAR
SHOES
LEVELING PLATES
BASE RING

Figure 7.54 Small Kingsbury six-shoe, two direction thrust bearing. Left-hand group assembled, except for one shoe and 'upper' leveling plate. Right-hand group disassembled (Courtesy of Kingsbury, Inc.)

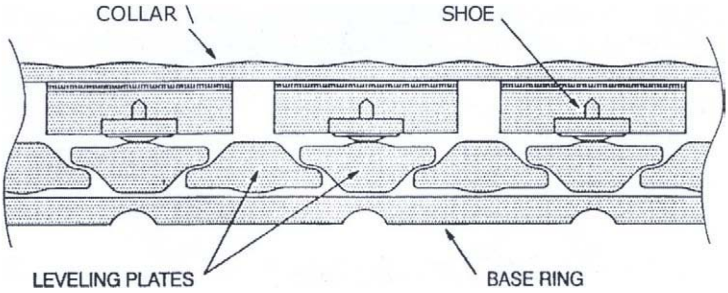

COLLAR
SHOE
LEVELING PLATES
BASE RING

Figure 7.55 Self-equalizing tilt-pad thrust bearing (View - looking down on assembly) (Courtesy of Kingsbury, Inc.)

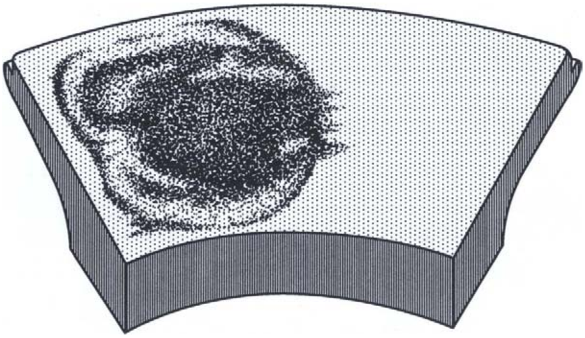


Figure 7.56 Evidence of overload on a tilt-pad self-equalizing thrust bearing pad (Courtesy of Kingsbury Corp.)

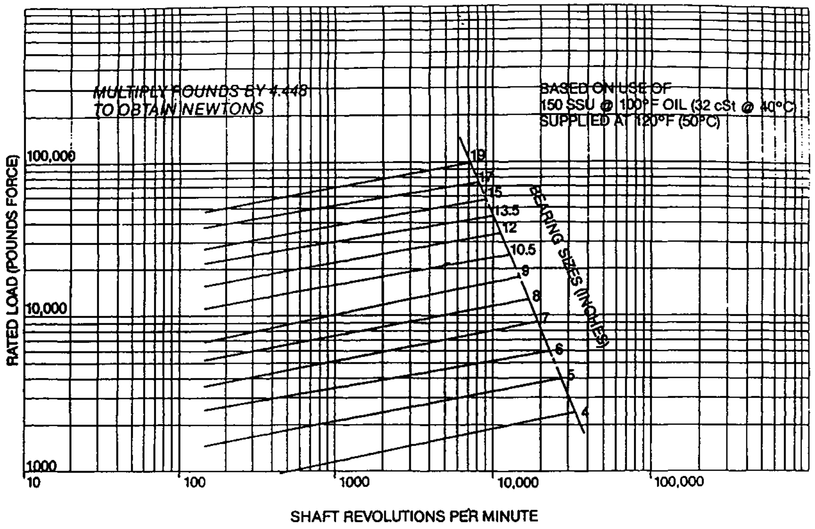


Figure 7.57 Thrust bearing rated load vs. speed (Courtesy of Kingsbury Corp.)

The oil film breaks down, thus allowing contact between the steel thrust collar and soft thrust bearing pad overlay (Babbitt). Once this thin layer (1/16") is worn away, steel to steel contact occurs resulting in significant turbo-compressor damage.

Thrust pad temperature sensors, located directly behind the babbitt at the pad maximum load point protect the compressor by tripping the unit before steel to steel contact can occur.

Figure 7.57 presents different Kingsbury bearing size rated capacities as a function of speed.

Figure 7.58 shows how thrust pad temperature and thrust load are related for a given thrust bearing size and shaft speed. Note that the greater the thrust load (P.S.I.), the smaller the oil film and the greater the effect of oil viscosity on oil flow and heat removal. Based on a maximum load of 500 psi, it can be seen from Figure 7.58 that a turbo-compressor thrust bearing pad temperature trip setting should be between 260° and 270°F.

Other than to support the rotor in an axial direction, the other function of the thrust bearing is to continuously maintain the axial position of the rotor. This is accomplished by locating stainless steel shims between the thrust bearing assembly and compressor axial bearing support plates. The most common thrust assembly clearance with the thrust shims installed is 0.011 – 0.014. These values vary with thrust bearing size. The vendor instruction book must be consulted to determine the proper clearance.

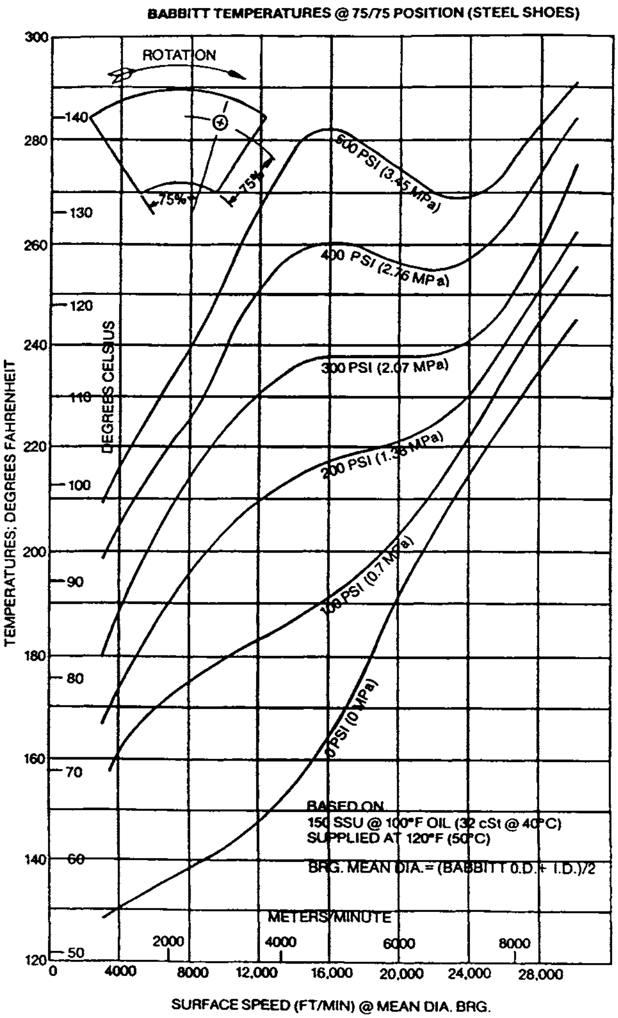


Figure 7.58 The relationship between thrust pad temperature and thrust load (Courtesy of Kingsbury, Inc.)

The following procedure is used to assure that the rotor is properly positioned in the axial direction.

1. With thrust shims removed, record total end float by pushing rotor axially in both directions (typically .250"–.500").
2. Position rotor as stated in instruction book.
3. Install minimum number of stainless steel thrust shims to limit end float to specified value.*

*An excessive number of thrust shims act as a spring resulting in a greater than specified axial clearance during full thrust load conditions.

Proper running position of the rotor is critical to obtaining optimum efficiency and preventing axial rubs during transient and upset conditions (start-up, surge, etc.)

Impeller thrust forces

Every reaction type compressor blade set or impeller produces an axial force towards the suction of the blade or impeller. Refer to Figure 7.59.

In this example, the net force towards the compressor suction is 2,000 psi for the set of conditions noted. Note that the pressure behind the impeller is essentially constant (50 psi), but the pressure on the front side of impeller varies (from 50 psi to 40 psi) because of the pressure drop across the eye labyrinth. Every impeller in a multistage compressor will produce a specific value of axial force towards it's suction at a specific flow rate, speed and gas composition. A change in any or all of these parameters will produce a corresponding change in impeller thrust.

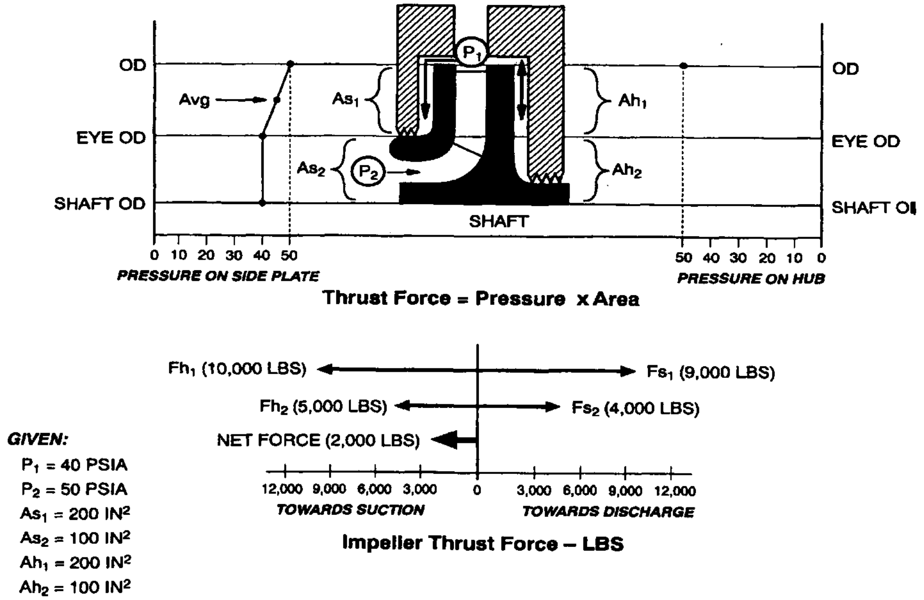
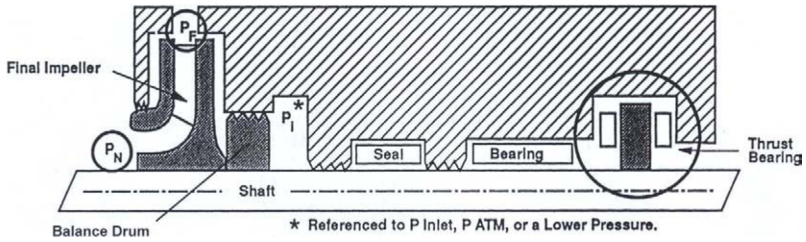

OD
EYE OD
SHAFT OD
Avg
As_1
As_2
P_1
P_2
Ah_1
Ah_2
SHAFT
OD
EYE OD
SHAFT O
0 10 20 30 40 50
PRESSURE ON SIDE PLATE
50 40 30 20 10 0
PRESSURE ON HUB
Thrust Force = Pressure x Area
Fh_1 (10,000 LBS)
Fs_1 (9,000 LBS)
Fh_2 (5,000 LBS)
Fs_2 (4,000 LBS)
NET FORCE (2,000 LBS)
12,000 9,000 6,000 3,000 0 3,000 6,000 9,000 12,000
TOWARDS SUCTION
TOWARDS DISCHARGE
Impeller Thrust Force – LBS
GIVEN:
P_1 = 40 PSIA
P_2 = 50 PSIA
As_1 = 200 IN^2
As_2 = 100 IN^2
Ah_1 = 200 IN^2
Ah_2 = 100 IN^2

Figure 7.59 Impeller thrust force


Final Impeller
P_F
P_I*
P_N
Seal
Bearing
Thrust Bearing
Shaft
Balance Drum
* Referenced to P Inlet, P ATM, or a Lower Pressure.

Total Impeller Thrust (LB) = $\sum$ Individual Impeller Thrust
Balance Drum Thrust (LB) = (P_F - P_I) x (Balance Drum Area)
Thrust Bearing Load (LB) = Total Impeller Thrust – Balance Drum Thrust

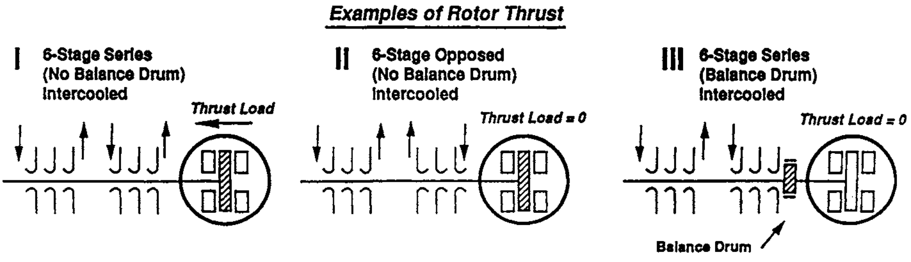

Examples of Rotor Thrust
I 6-Stage Series (No Balance Drum) Intercooled
Thrust Load
II 6-Stage Opposed (No Balance Drum) Intercooled
Thrust Load = 0
III 6-Stage Series (Balance Drum) Intercooled
Thrust Load = 0
Balance Drum

Figure 7.60 Rotor thrust force

Rotor thrust balance

Figure 7.60 shows how a balance drum or opposed impeller design reduces thrust force. The total impeller force is the sum of the forces from the individual impellers. If the suction side of the impellers is opposed, as noted in Figure 7.60, the thrust force will be significantly reduced and can approach 0. If the suction side of all impellers are the same (in series), the total impeller thrust force can be very high and may exceed the thrust bearing rating. If this is the case, a balance drum must be mounted on the rotor as shown in Figure 7.60. The balance drum face area is varied such that the opposing force generated by the balance drum reduces the thrust bearing load to an acceptable value. The opposing thrust force results from the differential between compressor discharge pressure (P_F) and compressor suction pressure (P_I) since the area behind the balance drum is usually referenced to the suction of the compressor. This is accomplished by a pipe that connects this chamber to the compressor suction. This line is typically called the 'balance line'.

It is very important to note that a balance drum is used only where the thrust bearing does not have sufficient capacity to absorb the total compressor axial load. And the effectiveness of the balance drum

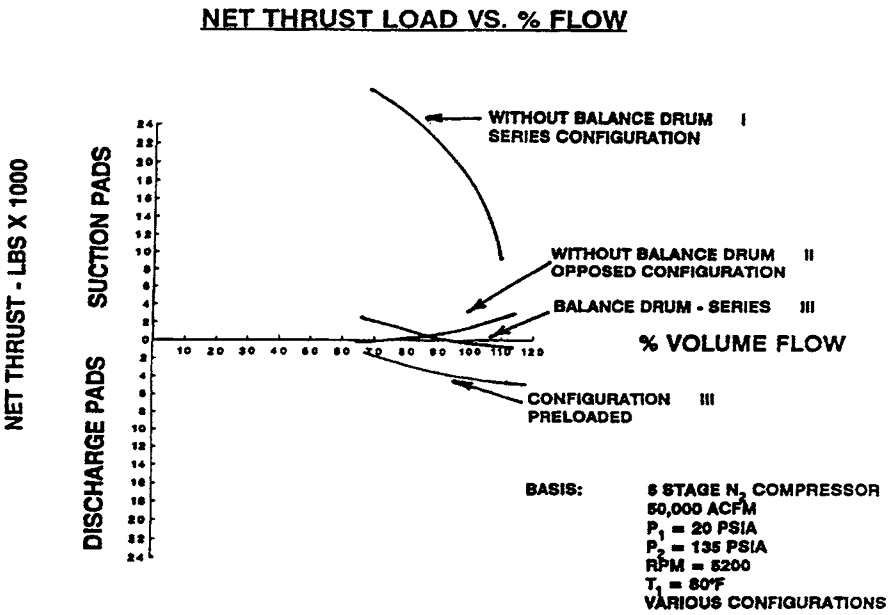


Figure 7.61 Rotor system designed four different ways

depends directly on the balance drum seal. Fail the seal, (open clearance significantly) and thrust bearing failure can result.

A common misunderstanding associated with balance drum systems is that a balance drum always reduces the rotor thrust to zero. Refer to Figure 7.61 and observe that this statement may or may not be true depending on the thrust balance system design. And even if it is, the thrust is zero only at one set of operating conditions.

Figure 7.61 shows a rotor system designed four (4) different ways. Note how the thrust **always** changes with the flow rate regardless of the design. Another misconception regarding thrust balance systems is the normal or 'active' direction of thrust. In many cases, the active thrust is assumed to always be towards the suction of the compressor. Observing Figure 7.61, it is obvious that the 'active' direction can change when the turbo-compressor has a balance drum or is an opposed design. It is recommended that the use of active thrust be avoided where possible and that axial displacement monitors be labeled to allow determination of the thrust direction at all times.

Please refer to Figure 7.62 which shows a typical thrust displacement monitor.

These monitors detect thrust position by targeting the shaft end, thrust collar or other collar on the rotor. Usually two or three probes (multiple voting arrangement) are provided to eliminate unnecessary compressor trips. The output of the probes is noted on the monitor as either + (normal) or – (counter). However, this information gives no direct indication of the axial direction of the thrust collar. The following procedure is recommended:

1. With compressor shutdown, push rotor towards the suction and note direction of displacement indicator.
2. Label indicator to show direction towards suction of compressor.

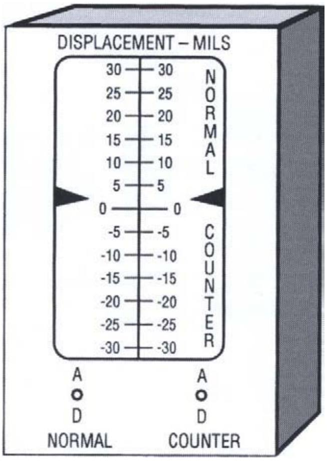


Figure 7.62 Typical axial thrust monitor

Knowing the actual direction of the thrust can be very useful during troubleshooting exercises in determining the root cause of thrust position changes.

Thrust condition monitoring

Failure of a thrust bearing can cause long term and possibly catastrophic damage to a turbo-compressor. Condition monitoring and trending of critical thrust bearing parameters will optimize turbo-compressor reliability.

The critical thrust bearing condition monitoring parameters are:

- Rotor position
- Thrust pad temperature
- Balance line ΔP

Rotor position is the most common thrust bearing condition parameter and provides useful information regarding the direction of thrust. It also provides an indication of thrust load but does not confirm that thrust load is high. Refer to Figure 7.63.

All axial displacement monitors have pre-set (adjustable) values for alarm and trip in both thrust directions. Typically, the established procedure is to record the thrust clearance (shims installed) during shutdown and set the alarm and trip settings as follows:

Alarm = $\frac{\text{Clearance}}{2}$ + 10 mils (each direction)

Trip = Alarm Setting + 5 mils (each direction)

The above procedure assumes the rotor is in the mid or zero position of the thrust clearance. An alternative method is to hand push the rotor to the assumed active position and add appropriate values for alarm and trip.

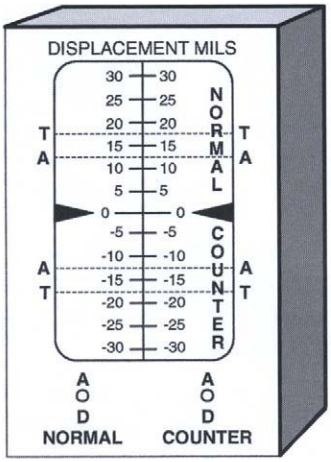
The writer personally recommends the first method since an active direction of thrust does not have to be assumed.

As noted, axial displacement monitors only indicate the quantity of thrust load. False indication of alarm or even trip settings can come from:

- Compression of thrust bearing components
- Thermal expansion of probe adaptors or bearing brackets
- Loose probes

It is strongly recommended that any alarm or trip displacement value be confirmed by thrust pad temperature if possible prior to taking action. Please refer back to Figure 7.58 of this chapter and note that the thrust pad temperature in the case of thrust pad overload is approximately 250°F. If an axial displacement alarm or trip signal is activated **observe** the corresponding thrust pad temperature. If it is below 220°F, take the following action:

- Observe thrust pads. If no evidence of high load is observed (pad



KEY:
A = ALARM
T = TRIP

NOTE: *Alarm and Trip Settings are Determined Based on Vendor's Recommendations*

Figure 7.63 Typical axial displacement monitor

and back of pad) confirm calibration of thrust monitor and change settings if necessary.

The last condition monitoring parameter for the thrust system is balance line pressure drop. An increase of balance line ΔP will indicate increased balance drum seal leakage and will result in higher thrust bearing load. Noting the baseline ΔP of the balance line and trending this parameter will provide valuable information as to the root cause of a thrust bearing failure. In many field case histories, the end user made many thrust bearing replacements until an excessive balance drum clearance was discovered as the root cause of the thrust bearing failure. It is a good practice to always check the balance line ΔP after reported machine surge. Surging will cause high internal gas temperatures which can damage the balance drum seal.

Mechanical seals

(Pump mechanical seal component knowledge)

- Introduction
- Function of mechanical seals
- The seal system
- Controlling flush flow to the seal
- Examining some causes of seal failures
- Seal configurations
- Flush system types
- Auxiliary stuffing box and flush plans

Introduction

The part of the pump that is exposed to the atmosphere and through which passes the rotating shaft or reciprocating rod is called the stuffing box. A properly sealed stuffing box prevents the escape of pumped liquid. Mechanical seals are commonly specified for centrifugal pump applications (Refer to Figure 7.64).

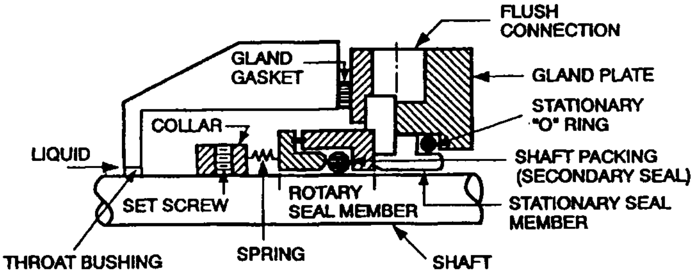

FLUSH CONNECTION
GLAND GASKET
GLAND PLATE
STATIONARY "O" RING
COLLAR
LIQUID
SHAFT PACKING (SECONDARY SEAL)
ROTARY SEAL MEMBER
SET SCREW
STATIONARY SEAL MEMBER
THROAT BUSHING
SPRING
SHAFT

Figure 7.64 Typical single mechanical seal

Function of mechanical seals

The mechanical seal is comprised of two basic components (Refer to Figure 7.65).

Basic seal components

- Stationary member fastened to the casing
- Rotating member fastened to shaft, either direct or with shaft sleeve

Figure 7.65 Basic seal components

The mating faces of each member perform the sealing. The mating surface of each component is highly polished and they are held in contact with a spring or bellows which results in a net face loading closure force (Refer to Figure 7.64).

In order to meet the objective of seal design (prevent fluid escape to the atmosphere), additional seals are required. These seals are either 'O' rings, gaskets or packing (Refer to Figure 7.64). For high temperature applications (above 400°) the secondary seal is usually 'Graphoil' or 'Kalrez' material in a 'U' or chevron configuration. An attractive alternative is to eliminate the secondary seal entirely by using a bellows seal since the bellows replaces the springs and forms a leak tight element thus eliminating the requirement for a secondary seal (Refer to Figure 7.66).

To achieve satisfactory seal performance for extended periods of time, proper lubrication and cooling is required. The lubricant, usually the pumped product, is injected into the seal chamber and a small amount passes through the interface of the mating surfaces. Therefore, it can be stated that all seals leak and the amount of leakage depends on the pressure drop across the faces. This performance can be considered like flow through an equivalent orifice (Refer to Figure 7.67).

The amount of heat generated at the seal face is a function of the face loading and friction coefficient, which is related to material selection and lubrication. Figure 7.68 shows the equation for calculating the amount of heat which needs to be removed by the flush liquid.

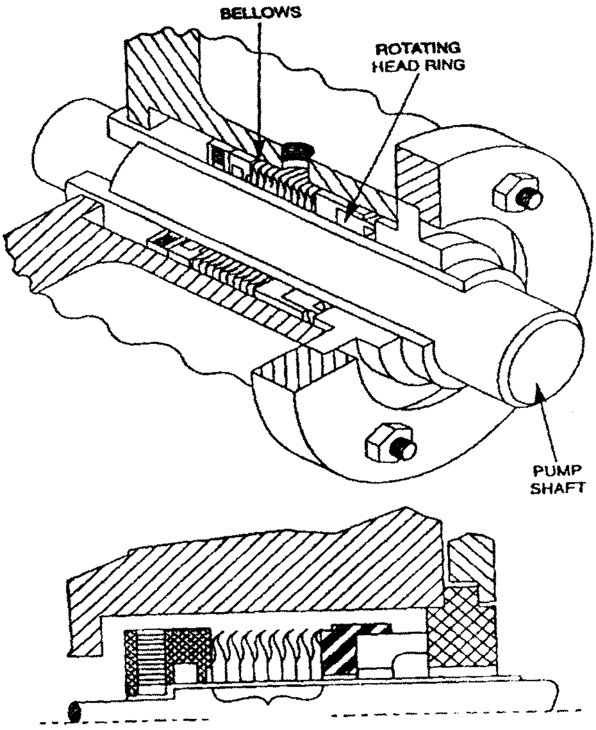

BELLOWS
ROTATING HEAD RING
PUMP SHAFT

Figure 7.66 Metal bellows seal

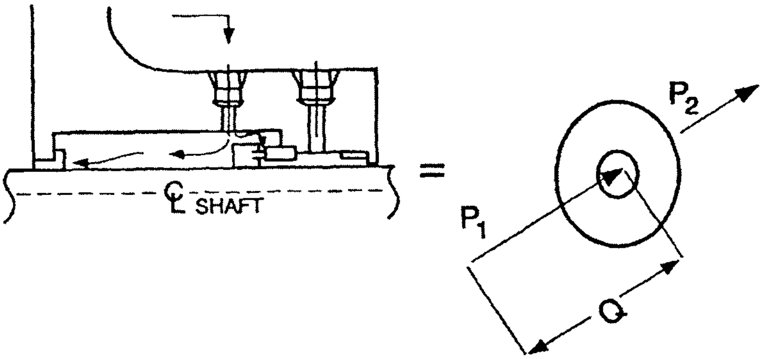

℄ SHAFT
=
P2
P1
Q

Figure 7.67 Equivalent orifice flow across seal faces

Heat generated by mechanical seal

$Q = 500 \bullet S.G. \bullet Q_{INJ} \bullet Cp \bullet \Delta T$

where: Q = heat load (BTU/HR)

$S.G.$ = specific gravity of injection liquid

C_p= specific heat of injection liquid $\frac{BTU}{LB\text{-}°F}$

ΔT = temperature rise of injection liquid (°F)

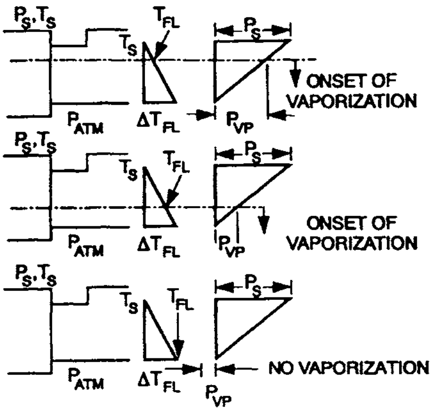
Q_{INJ}= injection liquid flow rate (G.P.M.)

Figure 7.68 Heat generated by mechanical seal

As the lubricant flows across the interface, it is prone to vaporization. The initiation point of this vaporization is dependent upon the flush liquid pressure and its relationship to the margin of liquid vapor pressure at the liquid temperature. The closer the liquid flush pressure is to the vapor pressure of the liquid at the temperature of the liquid, the sooner vaporization will occur (Refer to Figure 7.69).

The seal system

To assure reliable trouble free operation for extended periods of time, the seal must operate in a properly controlled environment. This requires that the seal be installed correctly so that the seal faces



P_S,T_S
T_FL
T_S
P_S
ONSET OF VAPORIZATION
P_ATM
ΔT_FL
P_VP
P_S,T_S
T_S
T_FL
P_S
ONSET OF VAPORIZATION
P_ATM
ΔT_FL
P_VP
P_S,T_S
T_S
T_FL
P_S
NO VAPORIZATION
P_ATM
ΔT_FL
P_VP

Figure 7.69 Typical seal face pressure temperature relationship to vaporization

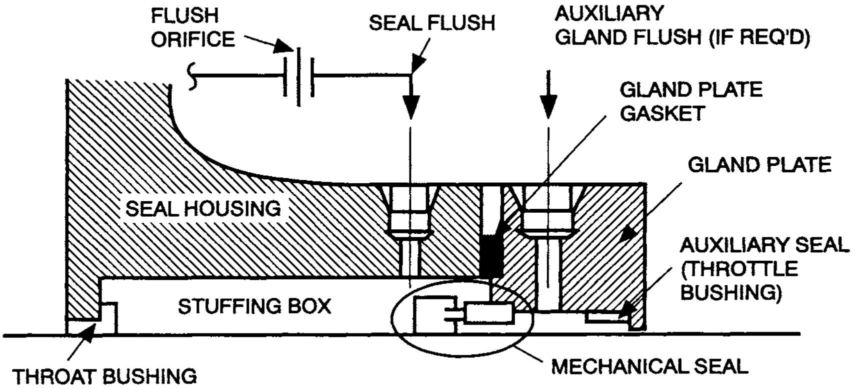

FLUSH
ORIFICE
SEAL FLUSH
AUXILIARY
GLAND FLUSH (IF REQ'D)
GLAND PLATE
GASKET
GLAND PLATE
SEAL HOUSING
AUXILIARY SEAL
(THROTTLE
BUSHING)
STUFFING BOX
THROAT BUSHING
MECHANICAL SEAL

Figure 7.70 Simple seal system

maintain perfect contact and alignment and that proper lubrication and cooling be provided. A typical seal system for a simple single mechanical seal is comprised of the seal, stuffing box throat bushing, liquid flush system, auxiliary seal and auxiliary flush or barrier fluid (when required) (Refer to Figure 7.70).

The purpose of the seal is to prevent leakage of pumped product from escaping to the atmosphere. The liquid flush (normally pumped product from the discharge) is injected into the seal chamber to provide lubrication and cooling. An auxiliary seal is sometimes fitted to the gland plate on the atmospheric side of the seal chamber. Its purpose is to create a secondary containment chamber when handling flammable or toxic fluids which would be considered a safety hazard to personnel if they were to leak to atmosphere. A liquid (non-toxic) flush or barrier fluid, complete with a liquid reservoir and appropriate alarm devices can be used to assure toxic fluid does not escape to the atmosphere.

Controlling flush flow to the seal

The simple seal system shown in Figure 7.71 incorporates an orifice in the flush line from the pump discharge to the mechanical seal. Its purpose is to limit the injection flow rate to the seal and to control pressure in the seal chamber. A minimum bore diameter or 1/8" is normally specified (to minimize potential of blockage) and the orifice can either be installed between flanges or in an orifice nipple.

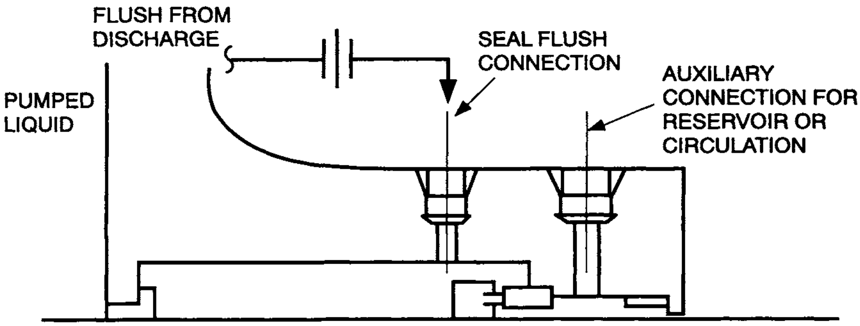


Figure 7.71 Seal flow control orifice

Examining some causes of seal failures

An indication of some causes of seal failures can be obtained while the seal is operating. When you consider the seal as an equivalent orifice, examination of 'tell tale' symptoms can indicate potential failure causes for which corrective action can be implemented or at least can provide direction of subsequent failure analysis (Refer to Figure 7.72). It should be noted that improper application, installation, and/or manufacturing errors can also result in mechanical seal failures.

Comments	Possible causes	Comments/recommendations
■ Seal squeal during operation	Insufficient amount of liquid to lubricate seal faces	Flush line may need to be enlarged and/or orifice size may need to be increased
■ Carbon dust accumulating on outside of seal area	Insufficient amount of liquid to lubricate seal faces	See above
	Liquid film vaporizing/ flashing between seal faces	Pressure in seal chamber may be too low for seal type
■ Seal spits and sputters in operation (popping)	Product vaporizing/flashing across seal faces	Corrective action is to provide proper liquid environment of the product at all times
		1. Increase seal chamber pressure if it can be achieved within operating parameters (maintain at a minimum of 25 Psig above suction pressure)

2. Check for proper seal balance with manufacturer
3. Change seal design to one not requiring as much product temperature margin (ΔT)
4. Seal flush line and/or orifice may have to be enlarged
5. Increase cooling of seal faces

Note: A review of seal balance requires accurate measurement of seal chamber pressure, temperature and product sample for vapor pressure determination

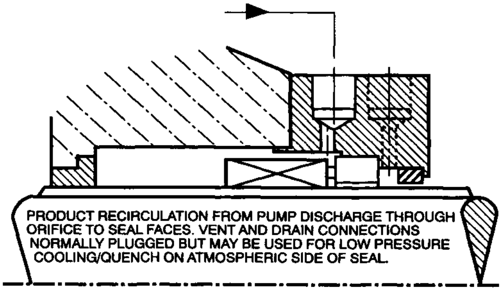
Figure 7.72 Possible causes of seal failure

Seal configurations

Mechanical seals are the predominant type of seals used today in centrifugal pumps. They are available in a variety of configurations, depending upon the application service conditions and/or the User's preference (Refer to Figures 7.73 to 7.76 for the most common arrangements used in refinery and petrochemical applications).

Single mechanical seal applications

Single mechanical seals (Refer to Figure 7.73) are the most widely used seal configuration and should be used in any application where the liquid- is non-toxic and non-flammable. As mentioned earlier in this section, many single mechanical seals are used with flammable and even toxic liquids and only rely on the auxiliary seal throttle bushing to prevent leakage to atmosphere. Since the throttle bushing does not positively contain leakage, State and Federal environmental regulations now require use of a tandem or double seal for these applications. In some plants, a dynamic type throttle bushing ('Impro' or equal) is used to virtually eliminate leakage of the pumped fluid to atmosphere in the event of a mechanical seal failure.


PRODUCT RECIRCULATION FROM PUMP DISCHARGE THROUGH ORIFICE TO SEAL FACES. VENT AND DRAIN CONNECTIONS NORMALLY PLUGGED BUT MAY BE USED FOR LOW PRESSURE COOLING/QUENCH ON ATMOSPHERIC SIDE OF SEAL

APPLICATIONS: NON HYDROCARBON, HYDROCARBON LIQUIDS. SPECIAL FEATURES INCORPORATED DEPENDING ON CHEMICAL CONTAMINANTS

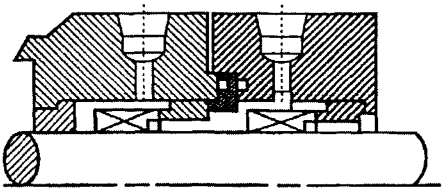
Figure 7.73 Single mechanical seal

Tandem mechanical seal applications

Tandem mechanical seals (Refer to Figure 7.74) are used in applications where the pumped fluid is toxic and/or flammable. They consist of two (2) mechanical seals (primary and back-up). The primary seal is flushed by any selected seal flush plan. The back-up seal is provided with a flush system incorporating a safe, low flash point liquid. A pressure alarm is provided to actuate on increasing stuffing box pressure between the primary and back-up seal thus indicating a primary seal failure. Since the pumped product now occupies the volume between the seals, failure of the back-up seal will result in leakage of the pumped fluid to atmosphere. In essence any time a tandem seal in alarm, it is actually a single seal and should be shut down immediately to assure that the toxic and/or flammable liquid does not leak to atmosphere.

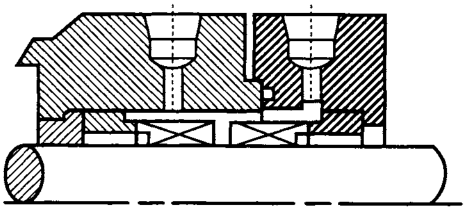
Double mechanical seal applications

Double mechanical seals (Refer to Figure 7.75) are used in applications where the pumped fluid is flammable or toxic and leakage to atmosphere cannot be tolerated under any circumstances. Typical process applications for double seals are H_2S service, Hydrofluoric acid alkylation services or sulfuric acid services.



APPLICATIONS: TOXIC, EXPLOSIVE HAZARD FROM LEAKAGE, CRYOGENIC LIQUIDS

Figure 7.74 Tandem seal arrangement



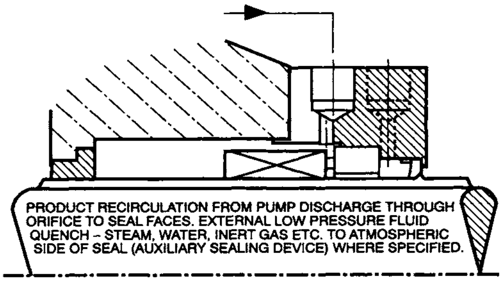
APPLICATIONS: SIMILAR TO THESE USED FOR TANDEM SEALS

Figure 7.75 Double mechanical seal

Leakage of the pumped fluid to the atmosphere is positively prevented by providing a seal system, whose liquid is compatible with the pumped liquid, that continuously provides a safe barrier liquid at a pressure higher than the pumped fluid. The seals are usually identical in design with the exception that one seal incorporates a pumping ring to provide a continuous flow of liquid to cool the seals. Typical double seal system components are: reservoir, cooler, pressure switch and control valve.

Liquid/Gas tandem mechanical seal applications

In this configuration (Refer to Figure 7.76) a conventional single liquid mechanical seal is used as the primary and a gas seal (non-contacting faces) that can temporarily act as a liquid seal in the event of primary seal failure serves as the back-up seal. This seal configuration is used in low specific gravity applications where the pumped fluid is easily vaporized. Using a gas seal as the back-up has the advantage of eliminating the vessel, cooler and pumping ring necessary for conventional tandem liquid seals.


PRODUCT RECIRCULATION FROM PUMP DISCHARGE THROUGH ORIFICE TO SEAL FACES. EXTERNAL LOW PRESSURE FLUID QUENCH - STEAM, WATER, INERT GAS ETC. TO ATMOSPHERIC SIDE OF SEAL (AUXILIARY SEALING DEVICE) WHERE SPECIFIED.

APPLICATIONS: CAN BE USED FOR LIQUIDS ABOVE AUTO IGNITION TEMP. TO PREVENT FIRE IF LEAKAGE EXPOSED TO ATMOSPHERE

Figure 7.76 Liquid/gas tandem seal combination

This application is well proven and has been used successfully for natural gas liquids, propane, ethylene, ethane and butane pump applications.

Double gas seal applications

Before leaving this subject, a relatively new application utilizes two (2) gas seals in a double seal configuration and uses N_2 or air as a buffer maintained at a higher pressure than the pumped fluid to positively prevent the leakage of pumped fluid to atmosphere. This configuration, like the tandem liquid/gas seal mentioned above eliminates the seal system required in a conventional liquid double seal arrangement. Note however, that the pumped product must be compatible with the small amount of gas introduced into the pumped fluid. This configuration cannot be used in recycle (closed loop) services.

An excellent resource for additional information covering design, selection and testing criteria, is API Standard 682.

Flush system types

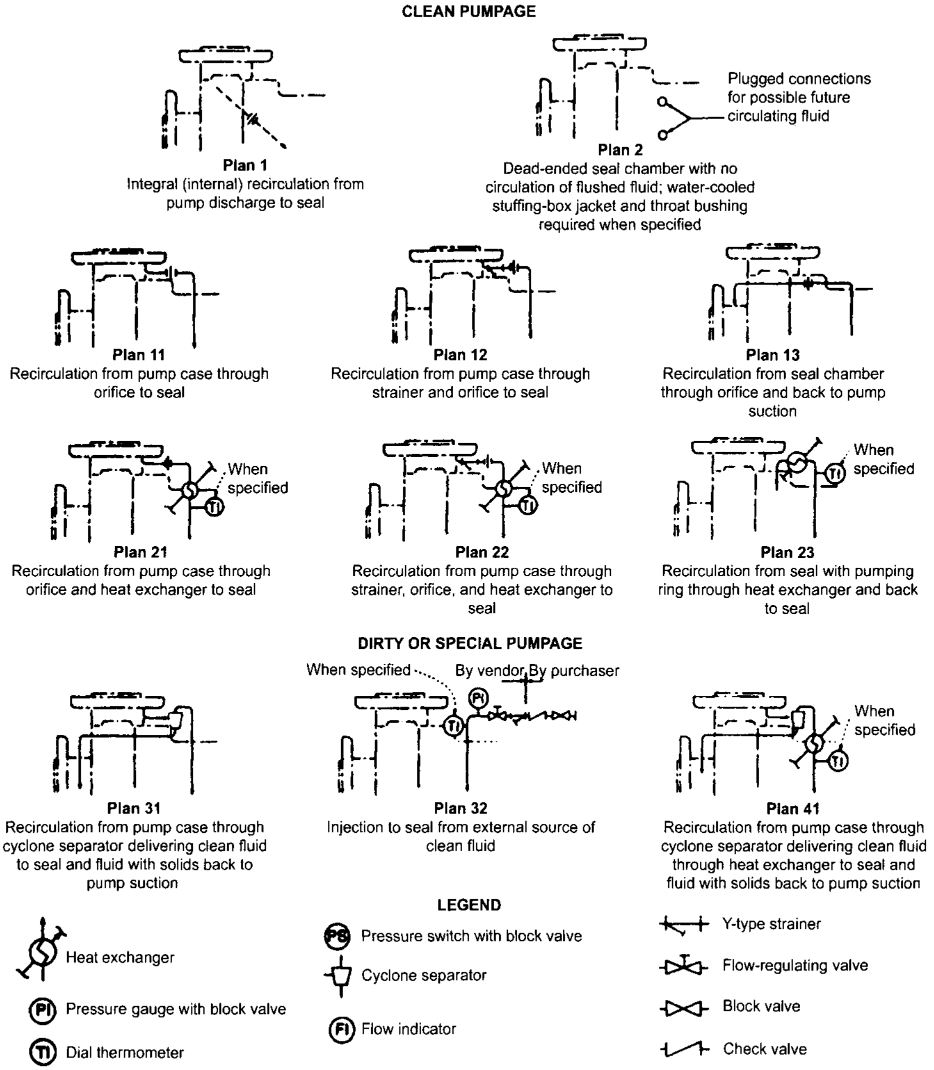
Providing the proper environment for the seal is a key factor in achieving satisfactory seal operation. In conjunction with defining the seal design and materials of construction, it is necessary to decide which type of seal flush system will be selected to lubricate and cool the seal faces. The API industry has developed various systems to accommodate the requirements of almost every possible sealing arrangement (Refer to Figure 7.77 and 7.77A).

Clean product systems (Plan 11)

In this plan, product is routed from the pump discharge to the seal chamber for lubricating and cooling the seal faces. It will also vent air and/or vapors from the chamber as it passes to the pump suction through the throat bushing. NOTE: For single stage pumps without back hub pumping vanes, it is necessary to have holes in the impeller that reduce the pressure behind the impeller below the discharge pressure to allow flush flow to exit the stuffing box through the throat bushing. (Refer to Figure 7.78).

Clean product flush (Plan 13)

In this plan, the product is routed from behind the pump impeller, through the throat bushing into the stuffing box and out of the stuffing box, through an orifice back to the suction (Refer to Figure 7.79) This plan is used primarily in vertical pump applications because it provides


CLEAN PUMPAGE
Plugged connections for possible future circulating fluid
When specified
When specified
When specified
DIRTY OR SPECIAL PUMPAGE
When specified
By vendor
By purchaser
PI
TI
When specified
LEGEND
Heat exchanger
Pressure gauge with block valve
Dial thermometer
Pressure switch with block valve
Cyclone separator
Flow indicator
Y-type strainer
Flow-regulating valve
Block valve
Check valve

Plan 1
Integral (internal) recirculation from pump discharge to seal

Plan 2
Dead-ended seal chamber with no circulation of flushed fluid; water-cooled stuffing-box jacket and throat bushing required when specified

Plan 11
Recirculation from pump case through orifice to seal

Plan 12
Recirculation from pump case through strainer and orifice to seal

Plan 13
Recirculation from seal chamber through orifice and back to pump suction

Plan 21
Recirculation from pump case through orifice and heat exchanger to seal

Plan 22
Recirculation from pump case through strainer, orifice, and heat exchanger to seal

Plan 23
Recirculation from seal with pumping ring through heat exchanger and back to seal

Plan 31
Recirculation from pump case through cyclone separator delivering clean fluid to seal and fluid with solids back to pump suction

Plan 32
Injection to seal from external source of clean fluid

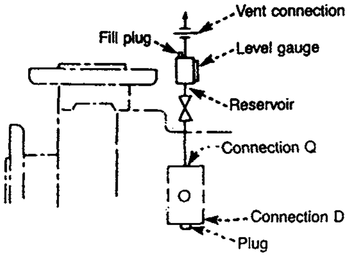
Plan 41
Recirculation from pump case through cyclone separator delivering clean fluid through heat exchanger to seal and fluid with solids back to pump suction

Figure 7.77 All flush plans reprinted with the permission of The American Petroleum Institute

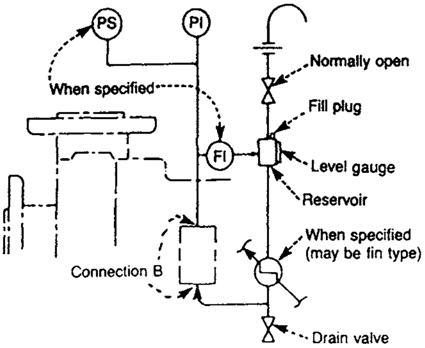
positive venting of the stuffing box. It is also used in single stage pump applications that do not employ pumping vanes or holes in the impellers.

Dirty product system (Plan 31)

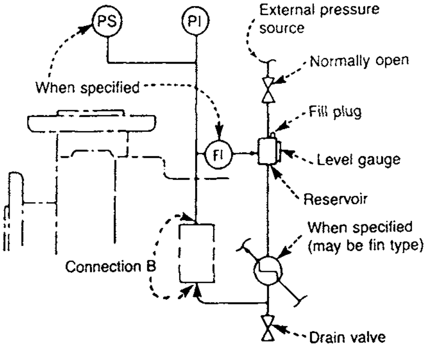
Liquid product from the pump discharge is routed to the seal chamber through a cyclone separator which is selected to optimize removal of solids across an individual pump stage (Refer to Figure 7.80 and to Figure 7.81 for guidelines covering the use of cyclones).


Vent connection
Fill plug
Level gauge
Reservoir
Connection Q
O
Connection D
Plug

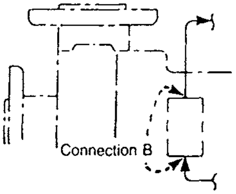
Plan 51
Dead-ended blanket (usually methanol, typically used with auxiliary sealing device (single- or double-seal arrangement


PS
PI
When specified
Normally open
Fill plug
FI
Level gauge
Reservoir
When specified (may be fin type)
Connection B
Drain valve

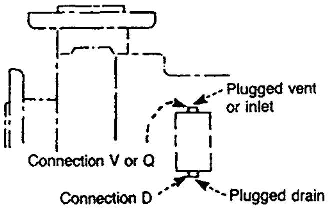
Plan 52
Nonpressurized external fluid reservoir with forced circulation; typically used with tandem-seal arrangement


PS
PI
External pressure source
When specified
Normally open
Fill plug
FI
Level gauge
Reservoir
When specified (may be fin type)
Connection B
Drain valve

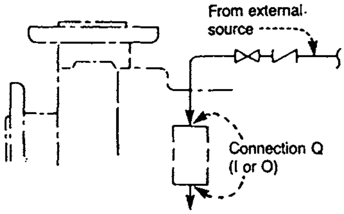
Plan 53
Pressurized external fluid reservoir with forced circulation; typically used with double-seal arrangement


Connection B

Plan 54
Circulation of clean fluid from external system typically used with double-seal arrangement

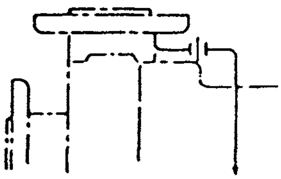

Plugged vent or inlet
Connection V or Q
Connection D
Plugged drain

Plan 61
Tapped connections for purchaser's use; applies when purchaser is to supply fluid (steam, gas, water, etc.) to auxiliary sealing device (single- or double-seal arrangement


From external source
Connection Q (I or O)

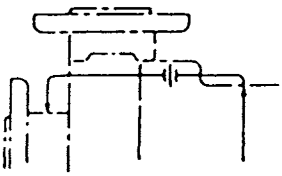
Plan 62
External fluid quench (steam, gas, water, etc.; typically used with throttle bushing or auxiliary sealing device (single- or double-seal arrangement

Figure 7.77A API Flush plans continued



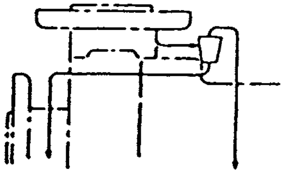
Plan 11
Recirculation from pump case through orifice to seal

Figure 7.78 API Flush plan 11



Plan 13
Recirculation from seal chamber through orifice and back to pump suction

Figure 7.79 API Flush plan 13



Plan 31
Recirculation from pump case through cyclone separator delivering clean fluid to seal and fluid with solids back to pump suction

Figure 7.80 API Flush plan 31

Guidelines for use of cyclones

- Do not use cyclones when differential pressure is less that 25 Psi
- Consider using orifice when pressure differential exceeds cyclone design differential
- Solids to be removed should have density at least twice that of the fluid
- Efficiency of separation is reduced as differential pressure across cyclone varies from design differential
- Separation efficiency drops as particle size decreases

Figure 7.81 Guidelines for use of cyclones

When a clean, cool seal flush liquid is mandated for reasons of preventing solids accumulation in the seal chamber, an external liquid flush system (Plan 32) is used. The fluid pressure is higher than that behind the impeller so that flow is always towards the pump suction,

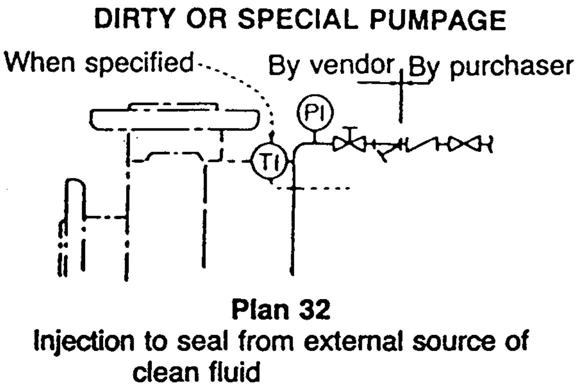


Figure 7.82 API Flush plan 32

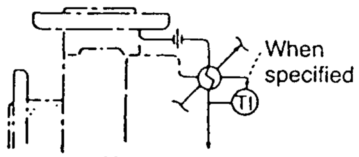
thus preventing back flow of dirty product into the seal chamber (Refer to Figure 7.82). When using Flush Plan 32, it is important to confirm that the flush fluid will not vaporize in the stuffing box.

High temperature product flush system (plan 23)

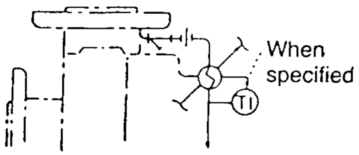
This flush plan is desirable when it is necessary to maintain the required margin between liquid vapor pressure (at seal chamber temperature) and seal chamber pressure. The feature about this plan is that the cooler only removes heat generated by the seal faces plus the heat soak through the shaft from the process. A throat bushing is installed in the seal chamber to isolate the product in this area from that in the impeller area of the pump. A circulating device (pumping ring) is mounted on the seal which circulates liquid in the seal chamber through a cooler and back to the seal chamber. It is more efficient than Plans 21 and 22 which incorporate a cooler to continuously cool the flush from the discharge. These plans are simply Flush Plan 1 1 with the addition of a cooler. Flush Plans 21, 22 or 23 should be used when the temperature of the pumped fluid is above 400°F and a bellows seal is not applied. It is important to confirm that the pumped fluid is clean when using Flush Plan 23 since there is not a constant external flush. This plan is typically used for boiler feed pump applications (Refer to Figure 7.83).

Low temperature product flush/buffer system (plan 52)

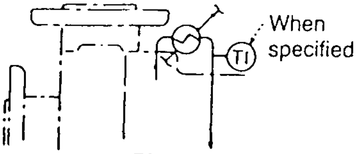
This system is well suited for low temperature applications such as ethylene, propylene and other low temperature liquids which are susceptible to forming ice on the seal faces when the atmospheric side of the seal is exposed to the atmosphere; thus separating the faces and resulting in excessive seal leakage. This plan consists of a tandem (dual) seal with a buffer liquid between them. A seal pot containing the buffer


When specified
TI

Plan 21
Recirculation from pump case through orifice and heat exchanger to seal

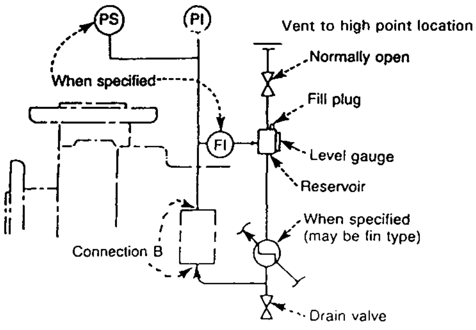

When specified
TI

Plan 22
Recirculation from pump case through strainer, orifice, and heat exchanger to seal


When specified
TI

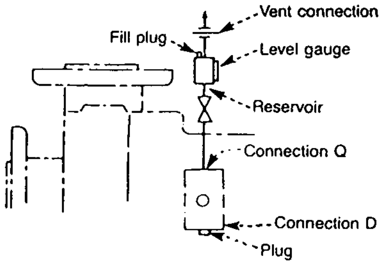
Plan 23
Recirculation from seal with pumping ring through heat exchanger and back to seal

Figure 7.83 API high temperature flush plans 21, 22, 23


PS
PI
When specified
Vent to high point location
Normally open
Fill plug
FI
Level gauge
Reservoir
When specified (may be fin type)
Connection B
Drain valve

Plan 52
Nonpressurized external fluid reservoir with forced circulation; typically used with tandem-seal arrangement

Figure 7.84A Plan 52

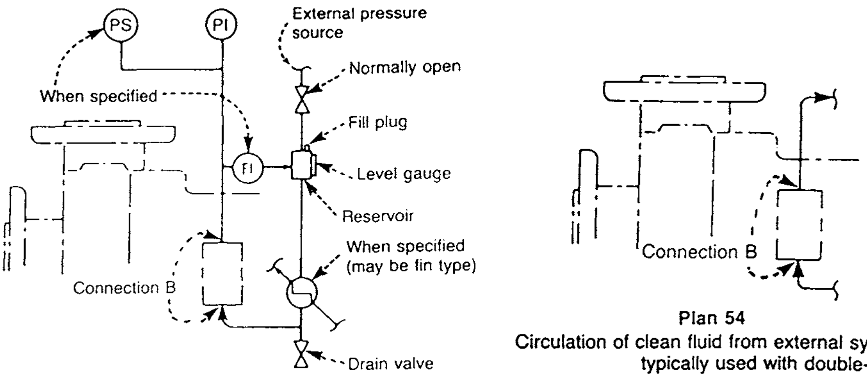

Vent connection
Fill plug
Level gauge
Reservoir
Connection Q
O
Connection D
Plug

Plan 51
Dead-ended blanket (usually methanol, see Note 3); typically used with auxiliary sealing device (single- or double-seal arrangement in Figure D-1)

Figure 7.84B Plan 51

liquid (usually methanol – a drying agent) is vented to a lower pressure vent system. The seal pot system is usually equipped with a pressure switch to sound an alarm if the inner seal product leakage cannot be adequately carried away through the orifice vent system (Refer to Figure 7.84A). When the alarm sounds, the pump should be shut down as soon as possible since the back-up seal is now functioning as the primary seal and will leak the pumped fluid to atmosphere if it fails.

Seal flush Pan 51 may also be used when leakage to atmosphere cannot be tolerated. Plan 51 incorporates a dead ended system design and is shown in Figure 7.84B.



Plan 53
Pressurized external fluid reservoir (see Note 3) with forced circulation; typically used with double-seal arrangement in Figure D-1

Plan 54
Circulation of clean fluid from external system typically used with double-seal arrangement

Figure 7.85A & B Plans 53 and 54

Toxic or flammable product flush system (plan 53)

This system is used when leakage to the atmosphere cannot be tolerated (see Figure 7.85A). It consists of a dual seal arrangement with a barrier liquid between them. A seal pot contains the barrier liquid at a pressure higher than seal chamber pressure (usually 20–25 PSI). Inner seal leakage will always be barrier liquid leakage into the product, resulting in some product contamination. The barrier liquid should be selected on the basis of its compatibility with the product. An internal pumping device (pumping ring) is used to circulate the barrier liquid into and out of the seal chamber through the seal pot. The integrity of the system always needs to be monitored to assure that seal pot pressure level is maintained with barrier liquid.

Plan 54 is also a dual system which utilizes a pressurized barrier liquid from an external reservoir or system to supply clean cool liquid to the seal chamber. As described in the previous plan, the barrier liquid pressure level is higher than the seal chamber pressure (usually 20–25 PSI) so that inner seal leakage is always into the pump. With this plan, it is also necessary to consider the compatibility of the barrier liquid and the pumped product. This system is considered to be one the most reliable systems available. However, it is more complex and more costly than other systems (Refer to Figure 7.85B).

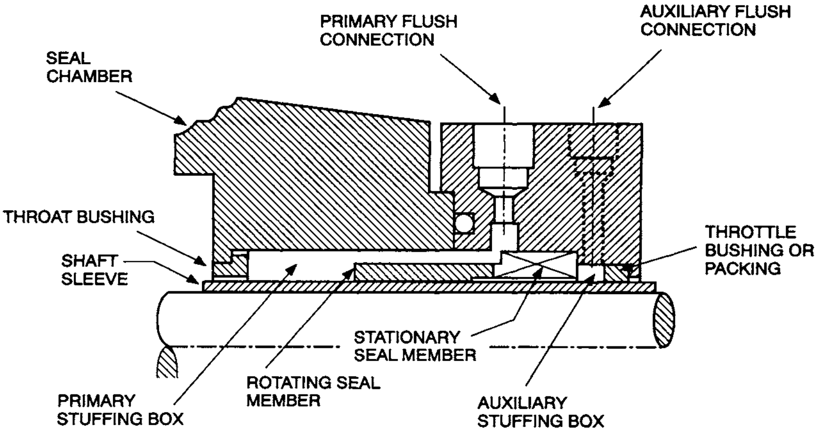

PRIMARY FLUSH CONNECTION
AUXILIARY FLUSH CONNECTION
SEAL CHAMBER
THROAT BUSHING
SHAFT SLEEVE
THROTTLE BUSHING OR PACKING
STATIONARY SEAL MEMBER
ROTATING SEAL MEMBER
PRIMARY STUFFING BOX
AUXILIARY STUFFING BOX

Figure 7.86 Auxiliary stuffing box

Auxiliary stuffing box and flush plans

As mentioned previously, the auxiliary stuffing box can be used as an auxiliary sealing device in the event of seal failure. (Refer to Figure 7.86). It is important to note that the auxiliary stuffing box contains a restricted flow seal (packing or close fitting throttle bushing) and does not positively contain the pumped fluid. Therefore, the auxiliary stuffing box seal device is for emergency containment of the pumped fluid only. The pump should be shut down immediately in the event of leakage observed from the auxiliary stuffing box if a quench is not supplied. It is always a good practice to require that the auxiliary stuffing box drain connection, which will come plugged, be piped to a drain system that meets environmental standards.

When the pumped fluid can vaporize and form hard deposits, the auxiliary stuffing box is used to contain a quench fluid that will dissolve (wash away) the hard deposits at the exit of the seal faces, thus eliminating seal face wear.

A typical refinery quench application is the use of low pressure (50 PSI) steam in and out of the auxiliary stuffing box to dissolve coke deposits on the seal face. This application also has the added advantage of keeping the standby pump seal warm which prevents thermal expansion problems in start-up. This arrangement uses a throttle bushing as the external seal.

Water is also used as an auxiliary stuffing box flush in caustic applications where solid deposits need to be flushed from the seal face. When a water flush is used, two (2) rows of packing are usually provided in the auxiliary stuffing box as an external seal to minimize external water leakage.

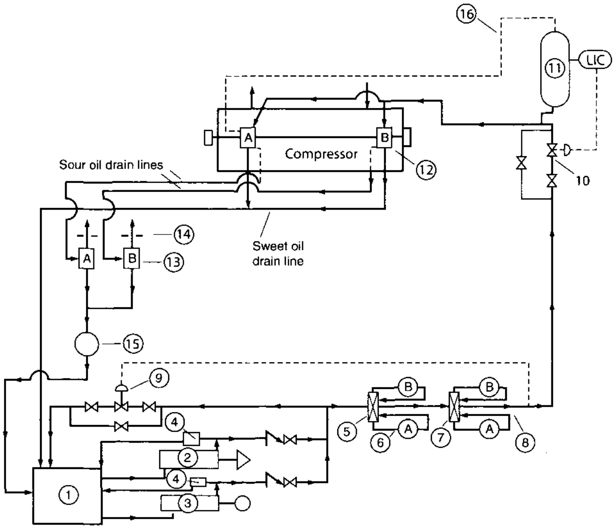
Compressor seal system overview and types
(Compressor liquid seal component knowledge)

- Introduction
- The supply system
- The seal housing system
- Seal supply systems
- Seal supply system summary
- Seal liquid leakage system

Introduction

There are numerous types of fluid seal systems since the types of seals utilized, sealing fluids and sealing pressures vary widely. Regardless of the type of seal used, the function of a critical equipment seal system is as follows: '*To continuously supply clean fluid to each specified seal interface point at the required differential pressure, temperature and flow rate*'. A typical seal system for a centrifugal compressor is shown in Figure 7.87.

The system shown is for use with clearance bushing seals. Let's examine



1. Oil reservoir
2. Main screw pump - turbine driven
3. Aux screw pump - motor driven
4. Relief valves
5. Transfer valve
6. Oil coolers (A & B)
7. Transfer valve
8. Oil filters (A & B)
9. Back pressure control valve
10. Level control valve
11. Overhead seal oil tank (typically 15 ft above compressor centerline)
12. ISO sleeve seals (also called liquid bushing seals)
13. Seal oil drainers (A & B)
14. Drainer orifice vents (A & B)
15. Degassing tank
16. Reference gas line to seal oil tank

Note: component condition instrumentation and autostarts not shown

Figure 7.87 API 614 lube/seal oil system for ISO-sleeve seals (Courtesy of Elliott Co.)

this figure by proceeding through the system from the seal oil reservoir through the compressor shaft seal and back through the reservoir. As previously discussed, the concept of sub-systems can be useful here. The seal oil system shown can be divided into four major sub-systems:

A The supply system

B The seal housing system

C The atmospheric drain system

D The seal leakage system

A The supply system

This system consists of the reservoir, pumping units, the exchangers, transfer valves, temperature control valves, and filters. The purpose of this sub-system is to continuously supply clean, cool sealing fluid to the seal interfaces at the correct differential pressure.

B The seal housing system

This system is comprised of two different seals. A gas side bushing, and an atmospheric bushing. The purpose of the seal housing system is to positively contain the fluid in the compressor and not allow leakage to the atmosphere. The seal fluid is introduced between both seal interfaces, thus constituting a double seal arrangement. Refer to Figure 7.88 for a closer examination of the seal.

The purpose of the gas side bushing seal is to constantly contain the reference fluid and minimize sour oil leakage. This bushing can be

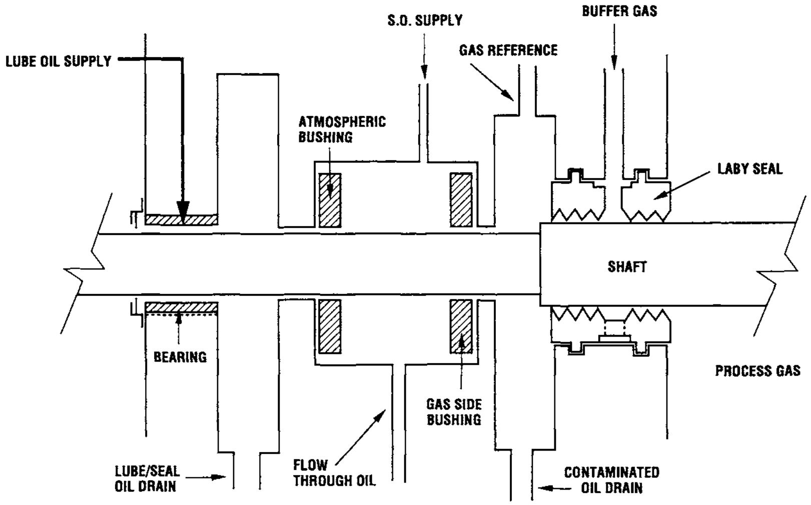

S.O. SUPPLY
BUFFER GAS
LUBE OIL SUPPLY
GAS REFERENCE
ATMOSPHERIC BUSHING
LABY SEAL
SHAFT
BEARING
PROCESS GAS
GAS SIDE BUSHING
LUBE/SEAL OIL DRAIN
FLOW THROUGH OIL
CONTAMINATED OIL DRAIN

Figure 7.88 Bushing seal schematic (Courtesy of M.E. Crane Consultant)

conceived as an equivalent orifice. This concept is similar to bearings previously discussed with the exception that the referenced downstream pressure of the gas side bushing can change. In order to assure a constant flow across this 'orifice,' the differential pressure must be maintained constant. Therefore, every compressor seal system is designed to maintain a constant differential against the gas side seal. The means of obtaining this objective will be discussed as we proceed.

The other seal in the system is the atmospheric bushing whose purpose is to minimize the flow of seal liquid to an amount that will remove frictional heat from the seal. This bushing can be conceptualized as a bearing, since the downstream pressure is usually atmospheric pressure. In systems that directly feed into a bearing the atmospheric bushing downstream pressure will be constant (approximately 20 psi). However, the upstream supply pressure will vary with the pressure required by the sealing media in the compressor.

As an example, if a seal system is designed to maintain a constant differential of 5 lbs. per square inch between the compressor process gas and the seal oil supply to the gas side bushing, the supply pressure with 0 PSIG process gas pressure, would be 5 psi to both the gas side bushing and atmospheric bushing. Therefore, gas side bushing and the atmospheric bushing differential would both be equal to 5 psi. If the process gas pressure were increased to 20 psi, the seal oil system would maintain a differential of 5 psi across the gas side seal, and the supply pressure to the gas side bushing and atmospheric bushing would be 25 psi. In this case, the differential across the gas side bushing would remain constant at 5 psi, but the atmospheric bushing differential pressure would increase from 5 to 25 psi. As a result, a primary concern in any seal liquid system is the assurance that the atmospheric bushing receives proper fluid flow under all conditions. After the seal fluid exits the seal chamber, it essentially returns through two additional subsystems.

C The atmospheric draining system

The flow from the atmospheric bushing, if it does not directly enter the bearing system, will return to the seal oil reservoir. In addition, flow from any downstream control valve will also return through the atmospheric drain system to the seal oil reservoir. Both these streams should be gas free since they should not come in contact with the process gas.

D The seal leakage system

The fluid that enters the gas side bushing is controlled to a minimum amount such that it can be either discarded or properly returned to the reservoir after it is degassed. Typically, this amount is limited to less than 20 gallons per day per seal. Since this liquid is in contact with the high speed shaft it is atomized and combines with sealing gas to enter the leakage system. This system consists of:

- An automatic drainer
- A vent system
- Degassing tank (if furnished)

The function of each component is as follows:

The drainer

The drainer contains the oil-gas mixture from the gas side seal. The liquid level under pressure in the drainer, is controlled by an internal float or external level control valve to drain oil back to the reservoir or the de-gassing tank, as required.

The vent system

The function of the seal oil drainer vent system is to assure that all gas side seal oil leakage is directed to the drainer. This is accomplished by referencing the drainer vent to a lower pressure than the pressure present at the gas side seal in the compressor. The drainer vent can be routed back to the compressor suction, suction vessel or a lower pressure source.

The degassing tank

This vessel is usually a heated tank, with ample residence time (72 hours or greater) to sufficiently de-gas all seal oil such that it will be returned to the reservoir and meet the seal oil specification. (Viscosity, flashpoint, dissolved gasses, etc.) These items will be discussed in detail later.

We will now proceed to discuss each of the major sub-systems in detail. Defining the function of each such that the total operation of a seal system can be simplified.

The supply system

Referring to definition of a seal oil system, it can be seen that the function is identical to that of a lube oil system, with one exception. The exception is that the seal fluid must be delivered to the seals at the specified differential pressure. Let's examine this requirement further.

Refer again to Figure 7.88 which shows an equivalent orifice diagram for a typical compressor shaft seal. Notice, that the atmospheric bushing downstream pressure is constant (atmospheric pressure). However, the gas side bushing pressure is referenced to the compressor process pressure. This pressure can and will vary during operation. If it were always constant, the requirement for differential pressure control would not be present in a seal system and would be identical to that of a lubricating system. Another way of visualizing the systems is to understand that the lube system utilizes differential pressure control as well, but the reference pressure (atmospheric pressure) is constant and consequently all control valves need only control lube oil pressure. However seal systems require some means of constant differential pressure control (reference gas pressure to seal oil supply pressure). This objective can be accomplished in many different ways. Referring back to Figure 7.87 it can be seen that the supply system function is identical to that of a lube oil system with the exception that the liquid is referenced to a pressure that can vary and must be controlled to maintain a constant differential between the referenced pressure and the seal system supply pressure. The sizing of the seal oil system components is also identical to that of the lube oil system components. Refer back and observe the heat load and flow required of each seal is determined in a similar way to that of the bearings. Seals are tested at various speeds and a necessary flow is determined to remove the heat of friction under various conditions. The seal oil flow requirements and corresponding heat loads, are then tabulated and pumps exchangers, filters, and control valves are sized accordingly.

The seal oil reservoir is sized exactly the same way as lube oil reservoir in our previous example. The only major difference between the component sizing of a seal and a bearing is that the seal flows across the atmospheric bushing will change with differential pressures. As previously explained, any liquid compressor seal incorporates a double seal arrangement. The gas side seal differential is held constant by system design. The atmospheric side seal differential varies with varying seal reference (process) pressure. Therefore, the total flow to the seals will vary with process pressure and must be specified for maximum and minimum values when sizing seal system components. Remembering the concept of an equivalent orifice, a compressor at atmospheric conditions will require significantly less seal oil flow than it will at high pressure (200 psi) conditions. This is true since the differential across the atmospheric seal and liquid flow will increase from a low value to a significantly higher value, while the gas side bushing differential and liquid flow will remain constant provided seal clearances remain constant.

Many seal system problems have been related to insufficient seal oil flow through the atmospheric bushing at low suction pressure conditions. Close attention to the atmospheric drain cavity temperature is recommended during any off design (low suction pressure condition) operation.

The seal housing system

Regardless of the type, the purpose of any seal is to contain the fluid in the prescribed vessel (pump, compressor, turbine, etc.). Types and designs of seals vary widely. Figure 7.89 shows a typical mechanical seal used for a pump.

Since the contained fluid is a liquid, this seal utilizes that fluid to remove the frictional heat of the seal and vaporize the liquid, thus attaining a perceived perfect seal. A small amount of vaporized liquid constantly exits the pump across the seal face. It is a fact that all seals leak. This is the major reason that many pump applications today are required to utilize seal-less pumps to prevent emission of toxic vapors. The following is a discussion of major types of seal combinations used in centrifugal compressor seal applications.

Gas seals

A typical gas seal is shown in Figure 7.90.

Gas seals have recently drawn attention since their supply systems appear to be much simpler than those of a traditional liquid seal system. Since gas seals utilize the sealed gas or a clean buffer gas, a liquid seal-system incorporating pumps, a reservoir and other components, is not required. However, one must remember that the sealing fluid still must be supplied at the proper flow rate, temperature and cleanliness. As a result, a highly efficient, reliable source of filtration, cooling, and supply must be furnished. If the system relies upon inert buffer gas for continued operation, the supply source of the buffer gas must be as reliable as the critical equipment itself. Gas seal configurations vary and will be discussed in detail in the next section. They can take the form of single, tandem (series), or multiple seal systems. The principle of operation is to maintain a fixed minimum clearance between the rotating and non-rotating face of the seal. The seal employed is

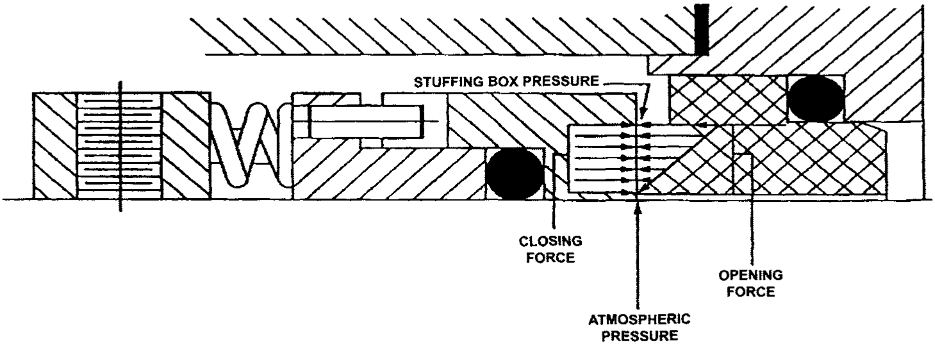
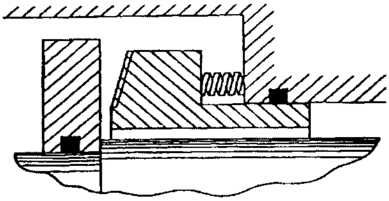
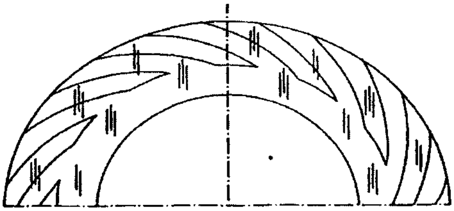


Figure 7.89 Typical pump single mechanical seal



Typical Design For Curved Face — Spiral Groove Non-contact Seal; Curvature May Alternately Be On Rotor



Typical Spiral Groove Pattern On Face Of Seal
Typical Non-contact Gas Seal

Figure 7.90 Typical gas seal (Courtesy of John Crane Co.)

essentially a contact seal with some type of lifting device to maintain a fixed minimum clearance between the rotating faces. It is essential that the gas between these surfaces be clean since any debris will quickly clog areas and reduce the effectiveness of the lifting devices, consequently resulting in rapid damage to the seal faces.

Liquid seals

Traditionally, the type of seal used in compressor service has been a liquid seal. Since the media that we are sealing against is a gas, a liquid must be introduced that will remove the frictional heat of the seal and assure proper sealing. Therefore, all compressor liquid seals take the form of a double seal. That is, they are comprised of two seals with the sealing liquid introduced between the sealing faces. Refer to Figure 7.91. To assure proper lubrication of both the gas side (inboard) and atmospheric side (outboard) seals, the equivalent ‘orifices’ of each seal must be properly designed such that the differential pressure present provides sufficient flow through the seal to remove the heat of friction at the maximum operating speed. The type of gas side seal used in Figure 7.91 is a contact seal similar to that used in most pump applications. This seal provides a minimum of leakage (five to ten gallons per day per seal) and provides reliable operation. (Continuous operation for 3+ years.) As will be discussed below, the specific types of seals used in the double seal (liquid) configuration can vary.

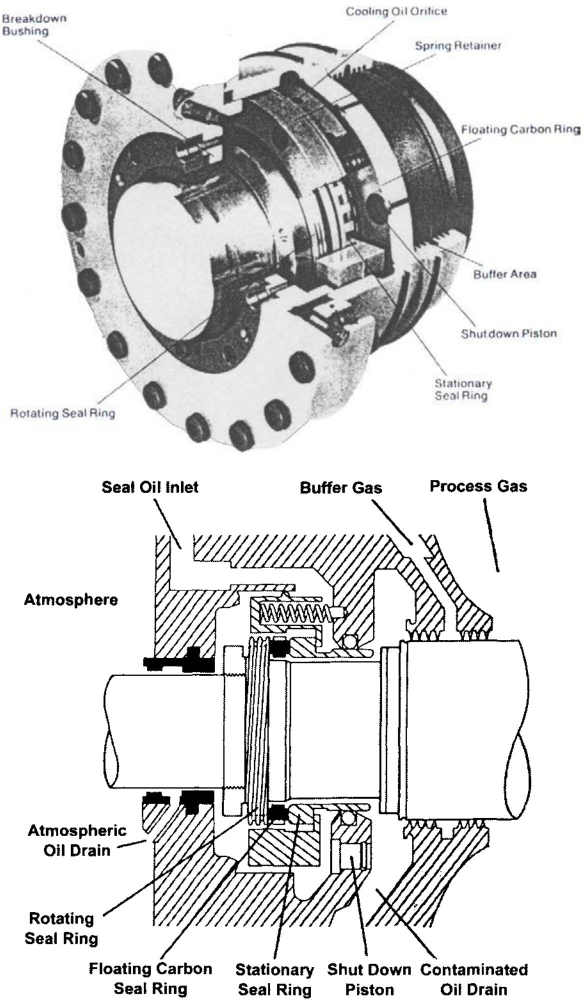

Breakdown Bushing
Cooling Oil Orifice
Spring Retainer
Floating Carbon Ring
Buffer Area
Shutdown Piston
Stationary Seal Ring
Rotating Seal Ring
Seal Oil Inlet
Buffer Gas
Process Gas
Atmosphere
Atmospheric Oil Drain
Rotating Seal Ring
Floating Carbon Seal Ring
Stationary Seal Ring
Shut Down Piston
Contaminated Oil Drain

Figure 7.91 Iso carbon seal (Courtesy of Elliott Co.)

Liquid bushing seals

A liquid bushing seal can be used for either a gas side or an atmospheric side seal application. Most seals utilize a liquid bushing seal for an atmospheric bushing application. A typical bushing seal is shown in Figure 7.92.

The principle of a bushing seal is that of an orifice. That is, a minimum clearance between the shaft and the bushing surface to minimize leakage. The bushing seal is designed such that the clearance is sufficient to remove all the frictional heat at the maximum power loss condition of that bushing with the available fluid differential across the

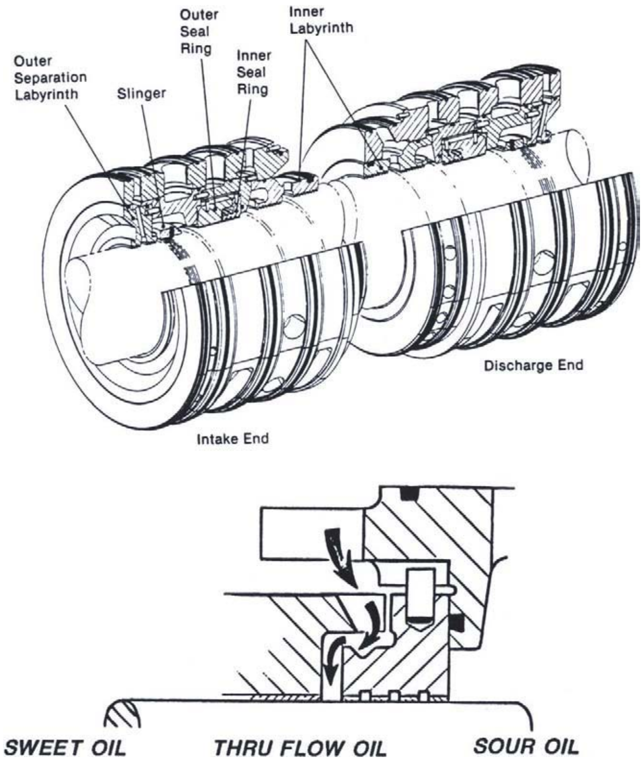


Figure 7.92 Bushing seal - top: oil film seal: bottom: seal oil flow (Courtesy of Dresser-Rand)

bushing. It is important to realize that while acting as a seal, the bushing must not act as a bearing. That is, it must have degrees of freedom (float) to assure that it does not support the load of the rotor. Since its configuration is similar to a bearing, if not allowed freedom of movement, it can act as an equipment bearing and result in a significant change to the dynamic characteristics of equipment with potential to cause damage to the critical equipment. In order to achieve the objectives of a bushing seal, clearances are on the order of 0.0005" diametrical clearance per inch of shaft diameter.

Liquid bushing seals are also used for gas side seals, however, their leakage rate will be significantly larger than that of a contact seal since they are essentially an orifice. When used as a gas side bushing, therefore, the system must be designed to minimize the differential across the bushing. As a result, the differential control system utilized must be accurate enough to maintain the specified oil/gas differential under all operating conditions. The typical design differential across a gas side bushing seal is on the order of five to ten psid. The accurate control of this differential is usually maintained by a level control system.

Referring back to Figure 7.92, one can see that functioning of the bushing seal totally depends on maintaining a liquid interface between the seal and shaft surface. Failure to achieve this results in leakage of gas outward through the seal. It must be fully understood that all bushing seals must continuously maintain this liquid interface to assure proper sealing. All systems incorporating gas side bushing seals must have the seal system in operation whenever pressurized gas is present inside the compressor case. If a liquid interface is not maintained, gas will migrate across the atmospheric bushing seal and proceed through the system returning back to the supply system. There have been cases in such system designs where failures to operate the seal system when the compressor is pressurized have resulted in effectively turning the gas side bushing into a filter for the entire process gas system! This resulted in the supply side of the seal oil system being filled with extensive debris that required lengthy flushing and system cleaning operations prior to putting the unit back into service. Remember, any system incorporating a gas side bushing seal must be designed such that the entrance of process gas into the supply system is prohibited at all time. This can be accomplished by either:

- Continuous buffer gas supply
- A check valve installed as close as possible to seals in the seal oil supply header
- Rapid venting and isolation of the compressor case on seal system failure

In the second and third cases above, supply seal oil piping must be thoroughly checked for debris prior to re-start of the compressor. It is our experience that many bushing seal system problems have resulted from improper attention to the above facts.

Contact seals

Figure 7.93 shows a typical compressor contact seal. As mentioned, these seals are similar in design to pump seals. In order to remove the heat of friction for this type of seal, a sufficient differential pressure above the reference gas must be maintained. Typical differentials for contact seals vary between 35 and 50 lbs. per square inch differential pressure. Leakage rates with a properly installed seal can be maintained between five to ten gallons per day per seal.

A limitation in the use of contact seals are shaft speed, since the contact seal operates on a surface perpendicular to the axis of rotation, the rubbing speed of the seal surface is critical. As a result, contact seals are speed limited. Typical maximum speeds are approximately 12,000 revolutions per minute. Above those speeds, bushing seals are used, since the sealing surface is maintained at a lower diameter and corres-

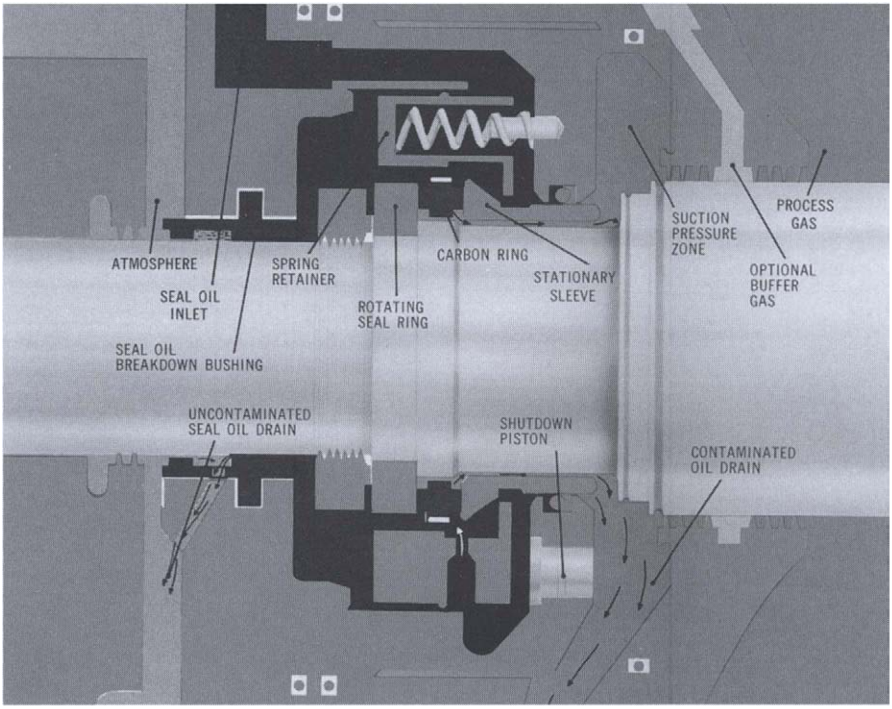

ATMOSPHERE
SEAL OIL INLET
SEAL OIL BREAKDOWN BUSHING
SPRING RETAINER
ROTATING SEAL RING
CARBON RING
STATIONARY SLEEVE
SUCTION PRESSURE ZONE
PROCESS GAS
OPTIONAL BUFFER GAS
UNCONTAMINATED SEAL OIL DRAIN
SHUTDOWN PISTON
CONTAMINATED OIL DRAIN

Figure 7.93 Compressor contact seal (Courtesy of Elliott Co.)

pondingly lower rubbing speed. The maximum limit of differential pressure across contact seals is controlled by the materials of construction and is approximately 200 lbs. per square inch differential. As a result, contact seals are usually used for gas side seal applications. They are very seldom utilized for atmospheric seal applications since they are differential pressure limited.

Since the differential pressure required across the seal face is relatively high as compared to a bushing seal, contact seals utilize differential pressure control as opposed to level control for most bushing seals. This fact will be discussed in the next section.

Restricted bushing seals

The last type of seal to be discussed is a restricted bushing type seal. This type of seal is shown in Figure 7.94.

This particular type of restricted bushing seal utilizes a small pumping ring in the opposite direction of bushing liquid flow to compensate for the relatively large leakage experienced with bushing seals by introducing an opposing pumping flow in the opposite direction. Seals of this type can be designed for practically zero flow leakages. However, it must be pointed out that in variable speed applications, the pumping

Compact Design — allows shorter bearing spans for higher critical speeds of the compressor rotor. **Sleeve** (impeller) with interference fit under bushing — protects shaft and simplifies assembly and disassembly. Requires only a jack/puller bolt ring. **Spacer** fit at initial assembly — no field fitting of parts.

ITEM	DESCRIPTION
1	Shaft
2	Impeller
3	Stator
4	Stepped Dual Bushing
5	Bushing Cage
6	Nut
7	Shear Ring
8	Oil/Gas Baffle
9	Spacer Ring

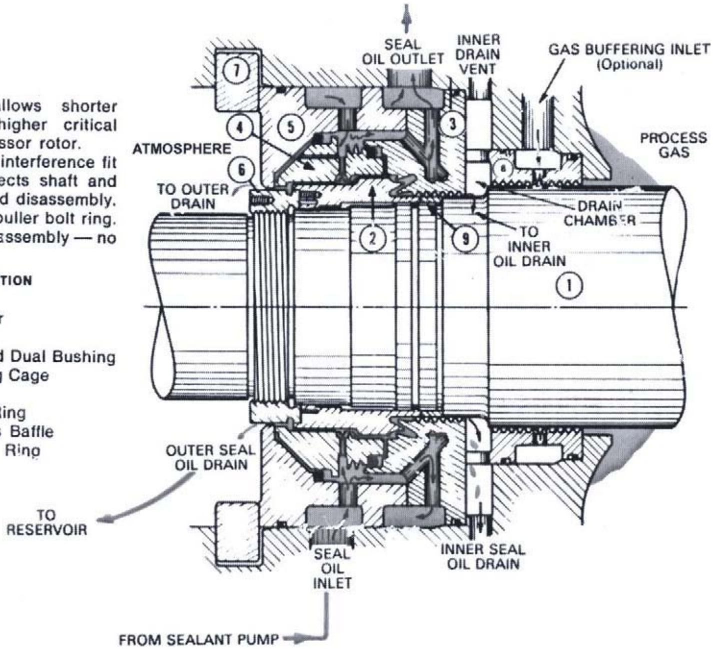

SEAL OIL OUTLET
INNER DRAIN VENT
GAS BUFFERING INLET (Optional)
PROCESS GAS
ATMOSPHERE
TO OUTER DRAIN
DRAIN CHAMBER
TO INNER OIL DRAIN
OUTER SEAL OIL DRAIN
TO RESERVOIR
SEAL OIL INLET
INNER SEAL OIL DRAIN
FROM SEALANT PUMP
1
2
3
4
5
6
7
8
9

Figure 7.94 Turbo-compressor 'trapped bushing seal' (Courtesy of A.C. Compressor Corp.)

capability of the trapped seal ring must be calculated for both minimum and maximum speeds. Failure to do so can result in the actual pumping of gas from the compressor into the sealing system. It is recommended that such seals be designed to leak a small amount at maximum operating speed. Any retrofits of equipment employing this type of seal should be investigated when higher operating speeds are anticipated. A restricted bushing seal is used exclusively for gas side service.

In summary, the basic types of liquid seals used for compressor applications can be either: open bushing types, contact types, or restricted bushing types. Contact types are used primarily on the gas side. Liquid bushing types are used on either the gas or atmospheric side. Restricted bushing types are used exclusively on the gas side.

We will now investigate various seal system designs using various seal combinations employing the types of seals that have been discussed in this section.

Seal supply systems

As can be seen from the previous discussion, the type of seal system will depend on the type of seal utilized. We will now examine five different types of seal systems, each utilizing a different type of main compressor shaft seal system. As we proceed through each type, the function of each system will become clear.

Example 1: Contact type gas side seal – bushing type atmospheric side seal with cooling flow

This system incorporates a contact seal on the gas side and a bushing seal on the atmospheric side of each end of the compressor. The inlet pressure of the seal fluid on each end is referenced to the suction pressure of the compressor. It should be noted that some applications employ different reference pressures on each end of the compressor. The reference pressure should be taken off the balance drum end, or high pressure end of the compressor, to assure that the oil to gas differential pressure is always at a minimum acceptable value. Therefore, the low pressure end may experience a slightly higher oil to gas differential then the reference end of the seal. Refer to Figure 7.95.

Proceeding through the seal, the seal oil supply, which is referenced to the gas reference pressure, enters the seal chamber. The differential across the gas side contact seal is maintained by a differential pressure

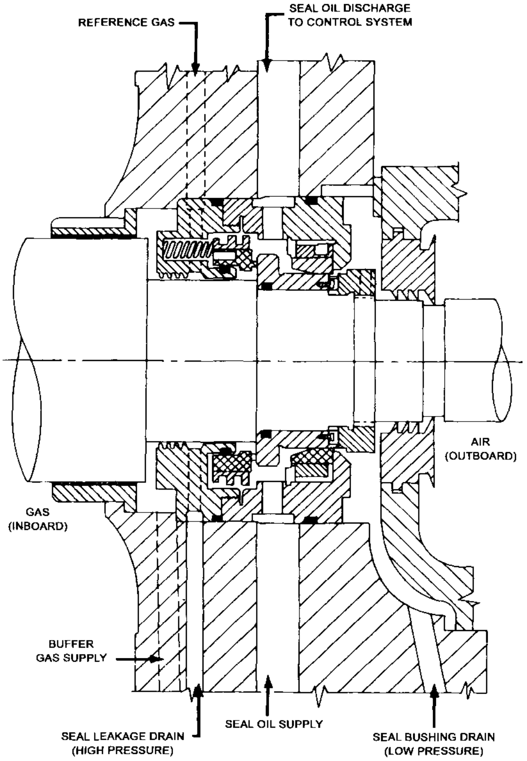

REFERENCE GAS
SEAL OIL DISCHARGE TO CONTROL SYSTEM
AIR (OUTBOARD)
GAS (INBOARD)
BUFFER GAS SUPPLY
SEAL LEAKAGE DRAIN (HIGH PRESSURE)
SEAL OIL SUPPLY
SEAL BUSHING DRAIN (LOW PRESSURE)

Figure 7.95 Compressor shaft seal (Courtesy of IMO Industries)

control valve located downstream of the seal. Seal oil flows in three separate directions:

- Through the seal chamber (cooling flow)
- Through the gas side contact seal (10–20 gallons/day)
- Through the atmospheric seal

Let's examine the variants of flows across the equivalent orifice of each portion of this configuration.

The gas side contact seal will experience a constant flow, that for purposes of discussion, can be assumed to be zero gallons per minute (since the maximum flow rate will usually be on the order of ten gallons per day).

The atmospheric side bushing seal flow will vary based upon the referenced gas pressure. At low suction pressure conditions, this flow will be significantly less than it will be under high pressure conditions. The seal system design must consider the maximum reference pressure to be experienced in the compressor case to assure that sufficient seal oil flow is available at maximum pressure conditions.

The seal chamber through flow in this seal design is used to remove any excess frictional heat of the seals and is regulatedby the downstream control valve. As an example, let us assume the following values were calculated for this specific seal application.

1. *Gas Side Seal Flow* = 0 GPM
2. *Atmospheric Side Seal Flow*
   Reference Pressure = 0 PSIG
   Seal Flow = 5 GPM
   Reference Pressure = 200 PSIG
   Seal flow = 12 GPM
3. *Flow Through Flow*
   Minimum = 3 GPM (occurring at high ATM bushing flow = 12 GPM)
   Maximum = 12 GPM (occurring at low ATM bushing flow = 3 GPM)
4. *Seal Oil Supply Flow* in both cases = 15 GPM

As shown in the previous example, the required seal oil supply at maximum operating speed required to remove frictional heat is 15 gallons a minute. At start-up, low suction pressure conditions, the control valve must open to allow an additional ten gallons a minute flow through to the seal chamber. At maximum operating pressure, however, the valve only passes a flow of three gallons a minute since 12 gallons a minute exit through the atmospheric bushing. This type of system is less sensitive to low suction pressure operation since flow through oil will remove frictional heat around the atmospheric bushing.

Example 2: Contact type gas side seal – bushing type atmospheric seal with orificed through flow

The only difference between this type of system and the previous system is that the back pressure is maintained constant by a permanently installed through flow orifice. As a result, the differential pressure control valve is installed on the inlet side of the system. The process gas reference is still the same as before, that is, to the highest pressure side of the compressor. Figure 7.96 shows this type of system.

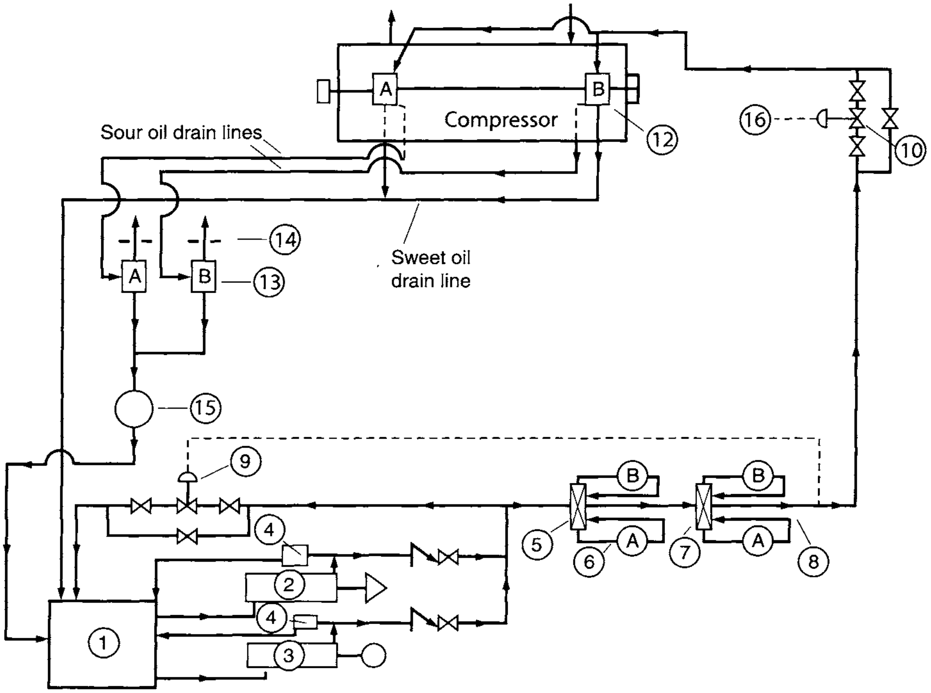
Let us examine the previous example case for this system and observe the differences.

1. *Gas Side Seal Flow = 0 GPM*
2. *Atmospheric Side Seal Flow*
   Reference Pressure = 0 PSIG
   Seal Flow = 5 GPM
   Reference Pressure = 200 PSIG
   Seal Flow = 12 GPM
3. *Flow Through Flow (Orifice)*
   Minimum = 0.5 GPM
   Maximum = 3 GPM

As can be seen, this system is more susceptible to high temperature atmospheric bushing conditions at low suction pressures and must be observed during such operation to assure integrity of the atmospheric bushing. In this system, the control valve will sense supply oil pressure to the seal chamber and control a constant set differential, approximately 35 psid, between the reference gas pressure and the supply pressure. If continued low pressure operation is anticipated with such a system, consideration should be given to a means of changing the minimum flow and maximum flow orifice for various operation points. Externally piped bypass orifices could be arranged such that a bypass line with a large orifice for minimum suction pressure conditions could be installed and opened during this operation. It is important to note, however, that the entire supply system must be designed for this flow condition and control valve must be sized properly to assure proper flow at this condition. In addition, the low pressure bypass line must be completely closed during normal high pressure operation.

Example 3: Bushing gas side seal – bushing atmospheric side seal with no flow through provision

Figure 7.87 shows this type of seal system. In this type of system, the differential control valve becomes a level control valve sensing differential from the level in an overhead tank and is positioned upstream of the unit. Both bushings can be easily conceived as equivalent orifices. The gas side bushing flow will remain constant


Compressor
A
B
Sour oil drain lines
Sweet oil drain line
⑫
⑯
⑩
⑭
⑬
⑮
⑨
④
②
③
①
⑤
⑥
⑦
⑧

1. Oil reservoir
2. Main screw pump - turbine driven
3. Aux screw pump - motor driven
4. Relief valves
5. Transfer valve
6. Oil coolers (A & B)
7. Transfer valve
8. Oil filters (A & B)
9. Back pressure control valve
10. Differential pressure control valve
11. Overhead seal oil tank (typically 15 ft above compressor centerline)
12. ISO sleeve seals (also called liquid bushing seals)
13. Seal oil drainers (A & B)
14. Drainer orifice vents (A & B)
15. Degassing tank
16. Reference gas line

Figure 7.96 API 614 Lube-seal oil system for ISO-carbon seals (Courtesy of Elliott Company)

regardless of differential. The atmospheric bushing flow will vary according to seal chamber to atmospheric pressure differential.

Therefore, the atmospheric bushing must be designed to pass a minimum flow at minimum pressure conditions that will remove frictional heat and thus prevent overheating and damage to the seal. Since a gas side bushing seal is utilized, a minimum differential across this orifice must be continuously maintained.

Utilizing the concept of head, the control of differential pressure across the inner seal is maintained by a column of liquid.

As an example, if the required gas side seal differential of oil to gas is 5 psid, by the liquid head equation:

$$\text{Head} = \frac{2.311 \times 5 \text{ psid}}{.85} = 13.6 \text{ ft.}$$

Therefore maintaining a liquid level of 13.6 ft. above the seal while referencing process gas pressure will assure a continuous 5 psid gas side bushing differential. In this configuration, the control valve which senses its signal from the level transmitter, will be sized to continuously supply the required flow to maintain a constant level in the overhead tack. As an example, consider the following system changes from start up to normal operation.

Seal System Flow

Item	Start-up condition	Normal operation
Compressor suction pressure	0 PSIG	200 PSIG
Overhead tank reference pressure	0 PSIG	200 PSIG
Gas side seal bushing flow	0 GPM	0 GPM
Atmospheric side seal bushing flow	5 GPM	12 GPM
Seal pump flow	20 GPM	20 GPM
Bypass valve flow	15 GPM	8 GPM

In this example a change from the start-up to operating condition will increase gas reference pressure on the liquid level in the overhead tank and would tend to push the level downward. Any movement of the level in the tank will result in an increasing signal to the level control valve to open, thus increasing the pressure (assuming a positive displacement pump) to the overhead tank and reestablishing the pre-set level.

In the above example, at 200 psi reference pressure, the bypass valve would close considerably. To increase the pressure supply of the seal oil from 5 psi to 205 psi, the difference of bypass flow through the valve (7 gpm) is equal to the increased flow through the atmospheric bushing at this higher differential pressure condition. Utilizing the concept of equivalent orifices, it can be seen that the additional differential pressure across the atmospheric bushing orifice is compensated for by reducing the effective orifice area of the bypass control valve. This is accomplished by sensing the level in the head tank and maintaining it at a constant value by opening the seal oil supply valve.

As in the case of the orificed through flow example above, this configuration is susceptible to high atmospheric bushing temperatures at low suction pressures and must be monitored during this condition. Repeated high temperatures during low suction pressure conditions should give consideration to re-sizing of atmospheric bushing clearances during the next available turnaround. The original equipment manufacturer should be consulted to assure correct bushing sizing and supply system capability.

Example 4: Gas side bushing seal – atmospheric side bushing seal with through flow design

Refer to Figure 7.97. The only difference between this system and the previous system is that a through flow option is added to allow sufficient flow through the system during changing pressure conditions. The bypass valve in the previous system is replaced in this system by a level control valve referenced from a head tank level transmitter, and is installed downstream of the seal chamber. This system functions in exactly the same way as the system in Example 1. The only difference being that a level control valve in this example replaces the differential control valve in the previous example. Both valves have the same function, that is, to control the differential in the seal chamber between the seal oil supply and the referenced gas pressure. A level control valve is utilized in this example, however, since a bushing seal requires a significantly lower differential between the seal oil supply pressure and the gas reference pressure.

Consider the following example. Assume that a differential control valve would be used as opposed to a level control valve for the system in Figure 7.97. For the start-up case, the differential control valve would have to maintain a differential of 5 psi over the reference gas. When the reference gas pressure were 0 psi, the oil upstream pressure to the valve would be approximately 5 psi. For the operating case, maintaining the same 5 psi differential, the upstream pressure across the valve would be approximately 205 lbs instead of 5 psi. Consequently, the valve position would change significantly, but still would have to control the differential accurately to maintain 5 psi. Reduction of this pressure in any amount below 5 psi could result in instantaneous bushing failure. However, if a level control valve were installed, the accuracy of the valve would be measured in inches of oil instead of psi. Any level control

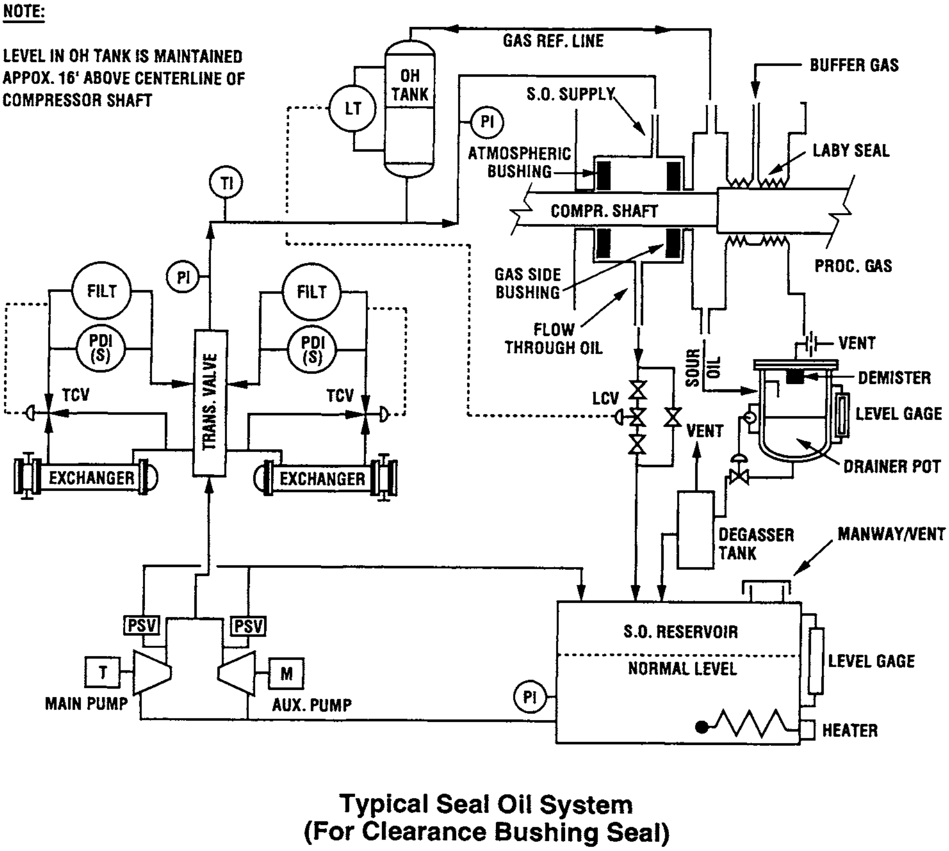

NOTE:
LEVEL IN OH TANK IS MAINTAINED APPOX. 16' ABOVE CENTERLINE OF COMPRESSOR SHAFT
GAS REF. LINE
BUFFER GAS
OH TANK
LT
S.O. SUPPLY
PI
LABY SEAL
ATMOSPHERIC BUSHING
TI
COMPR. SHAFT
GAS SIDE BUSHING
PROC. GAS
FILT
PI
FILT
FLOW THROUGH OIL
PDI (S)
PDI (S)
VENT
SOUR OIL
DEMISTER
TCV
TRANS. VALVE
TCV
LCV
LEVEL GAGE
VENT
EXCHANGER
EXCHANGER
DRAINER POT
DEGASSER TANK
MANWAY/VENT
PSV
PSV
S.O. RESERVOIR
NORMAL LEVEL
T
M
LEVEL GAGE
MAIN PUMP
AUX. PUMP
PI
HEATER
Typical Seal Oil System
(For Clearance Bushing Seal)

Figure 7.97 Typical seal oil system (Courtesy of M.E. Crane Consultant)

system could control the level within two inches, which would be only a .06 psid variation in pressure differential!

This example shows that the accurate means of controlling differential pressure for systems requiring control of small differential values, is to use level instead of differential control. This system would be designed such that the combination of the atmospheric flow and the through flow through the seal would be equal to the flow from the pump.

Example 5: Trapped bushing gas side seal: atmospheric side bushing seal with flowthrough design

This system would follow exactly the same design as the system described in Example 4. The only difference would be in the amount of flow registered in the seal oil drainer. A trapped bushing system is designed to minimize seal oil drainer pot leakage. Typical values can be less than five gallons/day.

Seal supply system summary

All of the above examples have dealt with a system incorporating one seal assembly. It must be understood that most systems utilize two or more seal system assemblies. Typical multi-stage compressors contain two seal assemblies per compressor body and many applications contain upwards of three compressor bodies in series, or six seal assemblies. Usually each compressor body is maintained at the suction pressure to that body, therefore three discreet seal pressure levels would be required and three differential pressure systems would be utilized. The concepts discussed in this section follow through regardless of the amount of seals in the system. Sometimes, the entire train, that is, all the seals referenced to the same pressure. In this case, one differential seal system could be used across all seals.

In conclusion, remembering the concept of an orifice will help in understanding the operation of these systems. Remember, the gas side bushing is essentially zero flow, the atmospheric side bushing flow varies with changing differential across the seal and any seal chamber through flow will change either as a result of differential across a fixed orifice or the repositioning of the control valve.

Seal liquid leakage system

This seal system sub-system's function is to collect all of the leakage from the gas side seal and return it to the seal reservoir at specified seal fluid conditions. Depending upon the gas condition in the case, this objective may or may not be possible. If the gas being compressed has a tendency to change the specification of the seal oil to off specification conditions, one of two possibilities remain:

- *Introduce a clean buffer gas* between the seal to assure proper oil conditions
- *Dispose of the seal oil leakage*

In most cases, the first alternative is utilized. Once the seal oil is in the drainer, a combination of oil and gas are present. A vent may be installed in the drainer pot to remove some of the gas, or a degassing tank can be incorporated.

This concludes the overview section of seal oil systems. As can be seen from the above discussion, it is evident that the design of a seal oil system follows closely to that of a lube system. The major difference is that the downstream reference pressure of the components (seal) varies, whereas in the case of a lube system it does not. In addition, the collection of the expensive seal oil is required in most cases and a downstream collector, or drainer system, must be utilized. Other than these two exceptions, the design of the seal system is very similar to that of a lube system and the same concepts apply in both cases.

Seal system component design and preventive maintenance

(Compressor liquid seal component knowledge)

- Introduction
- High pressure systems
- Differential pressure control valves
- Level control valves
- Control valve location in the system
- Seal system component design and preventive maintenance

Introduction

In this chapter we will deal with liquid seal supply system component designs of three major areas:

- The pumping system
- The differential control valves
- The level control valves

In the next chapter, we will deal with the downstream side of the seal system, the drainers, demisters, and degassing tanks.

There is a great similarity between the seal system and the lube system. As far as the components are concerned, they are similar. The reservoir, pumps, coolers, filters and control valves are sized in exactly the same way as those of a lube system, the only major areas of differences between a lube and seal supply systems are:

- The arrangement of components in high pressure systems
- Differential control

High pressure systems

High pressure seal systems are defined in general as those systems where the seal pressure is in excess of 1,000 lbs per square inch. This category of sealing systems experiences unique problems. System cooling requirements, high pressure pump start-up, and rapid deterioration of system components in the presence of debris. Refer to

Figure 7.98. This is a typical component arrangement for a combined lube and seal system.

System cooling requirements

In this arrangement, excess (bypass) oil is recirculated uncooled back to the reservoir. The temperature rise of the oil through the pump is proportional to pump differential pressure. In high pressure systems, a significant pump temperature rise occurs. If the combined effect of temperature rise and bypass flow are such that the heat input into the reservoir exceeds the heat loss in the reservoir, a reduction of oil viscosity will occur. High pressure positive displacement rotary pumps are sensitive to lower oil viscosities. At a given pressure, there is a minimum specified operating viscosity. For high pressure systems, careful attention must be drawn to location of the seal oil cooler. In many cases, a bypass cooler or a cooler at the suction of the pump, is recommended to preclude the possibility of pump damage.

Consideration should also be given to cooler leaks in high pressure systems. A small leak can result in a large quantity of oil being taken from the system in a short period of time. Coolers should be considered to be installed in the lowest possible pressure section of the system. In combined lube and seal systems, a recommendation is to install the cooler in the low pressure section of the system and utilize a booster pump for seal oil supply.

High pressure start

In high pressure systems, consideration must be given to the sudden starting of a seal pump against a high system pressure. Pumps can be quickly damaged under this condition. If a pilot operated relief valve is utilized, a time delay can be incorporated to hold the valve open upon simultaneously starting of the standby pump. This will alleviate any high pressure sudden shock problems with positive displacement pumps.

Component failure

The failure rate of components in high pressure systems is greater than other types of auxiliary systems. Pumps are particularly susceptible to failure. High pressure seal pumps can fail from high pressure hydraulic shocks induced during start up. As a result, a significant amount of debris is introduced into the system and the corresponding high pump discharge pressure will cause the relief valve to lift. This action will start the second pump. Since the second pump is usually connected to the same system, this pump will also experience high discharge pressure and recirculate all of its flow. This action will quickly result in high suction temperature in both pumps and dual pump failure, resulting in critical

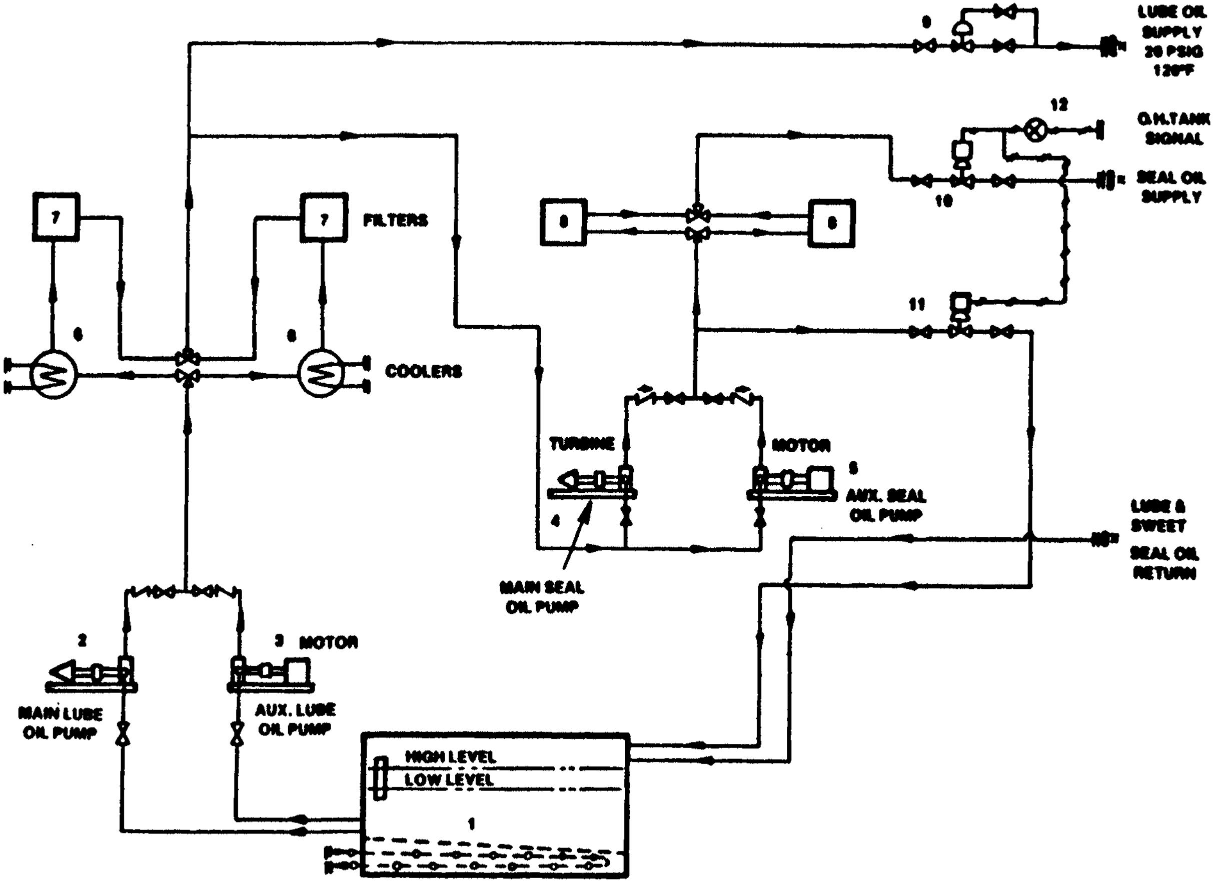

9
LUBE OIL SUPPLY 20 PSIG 120°F
12
O.H.TANK SIGNAL
SEAL OIL SUPPLY
10
7
7
FILTERS
8
8
11
6
6
COOLERS
TURBINE
MOTOR
5
AUX. SEAL OIL PUMP
4
LUBE & SWEET SEAL OIL RETURN
MAIN SEAL OIL PUMP
2
3
MOTOR
MAIN LUBE OIL PUMP
AUX. LUBE OIL PUMP
HIGH LEVEL
LOW LEVEL
1

Figure 7.98 High pressure, combined lube and seal system sealing pressurs to 3000 psig (211 Kg/CM^2) (Courtesy of Dresser Rand)

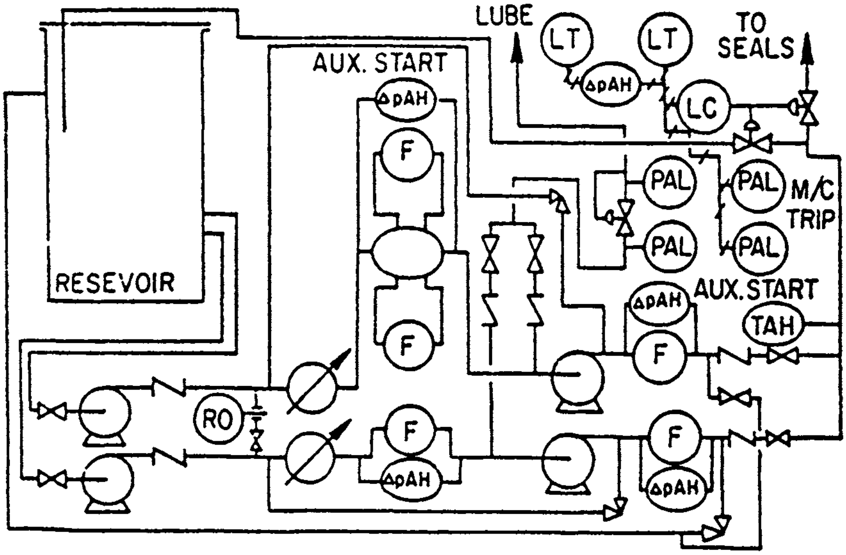


Figure 7.99 High pressure seal system schematic

equipment shutdown. Consideration should be given to separate pump loops as shown in Figure 7.99 for high pressure systems to preclude the possibility of dual pump failure.

Differential pressure control valves

Function

A differential pressure control valve is required in a sealing system since the referenced pressure or the gas side seal downstream pressure, does not remain constant. In order to supply a constant flow across the seal (which is an equivalent orifice) the pressure drop must remain constant. This is facilitated by using a differential pressure control valve, either direct operated as shown in Figure 7.100 or controller operated as shown in Figure 7.101.

The set point differential pressure is determined by the seal design. The required oil to gas differential pressure is that amount necessary to introduce sufficient flow to remove frictional heat from the seal. Refer to Figure 7.102 which shows a differential bypass control valve installed in the seal oil system. Let's examine valve operation during start-up and normal operating conditions.

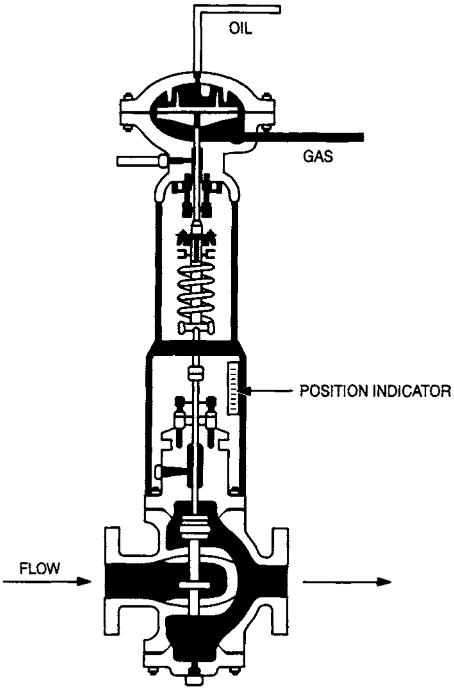

OIL
GAS
POSITION INDICATOR
FLOW

Figure 7.100 Direct acting differential pressure control valve

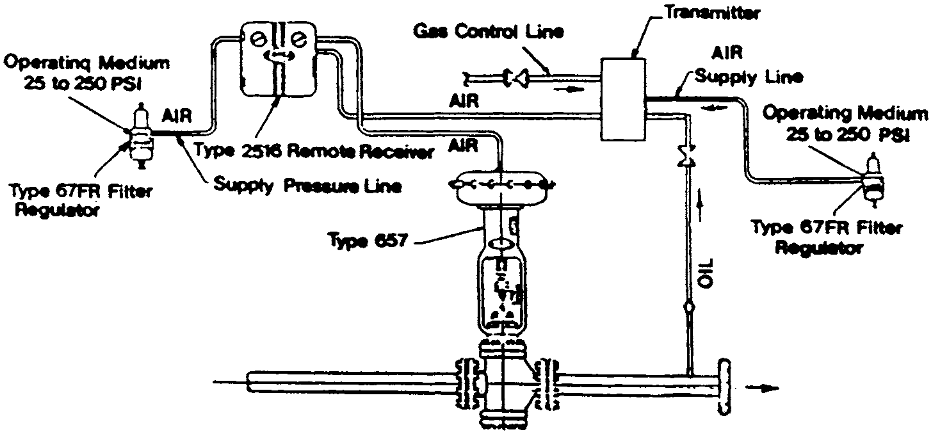

Transmitter
Gas Control Line
AIR
Supply Line
Operating Medium
25 to 250 PSI
AIR
Operating Medium
25 to 250 PSI
AIR
Type 2516 Remote Receiver
AIR
Supply Pressure Line
Type 67FR Filter
Regulator
Type 657
Type 67FR Filter
Regulator
OIL

Figure 7.101 Controller operated pressure control valve

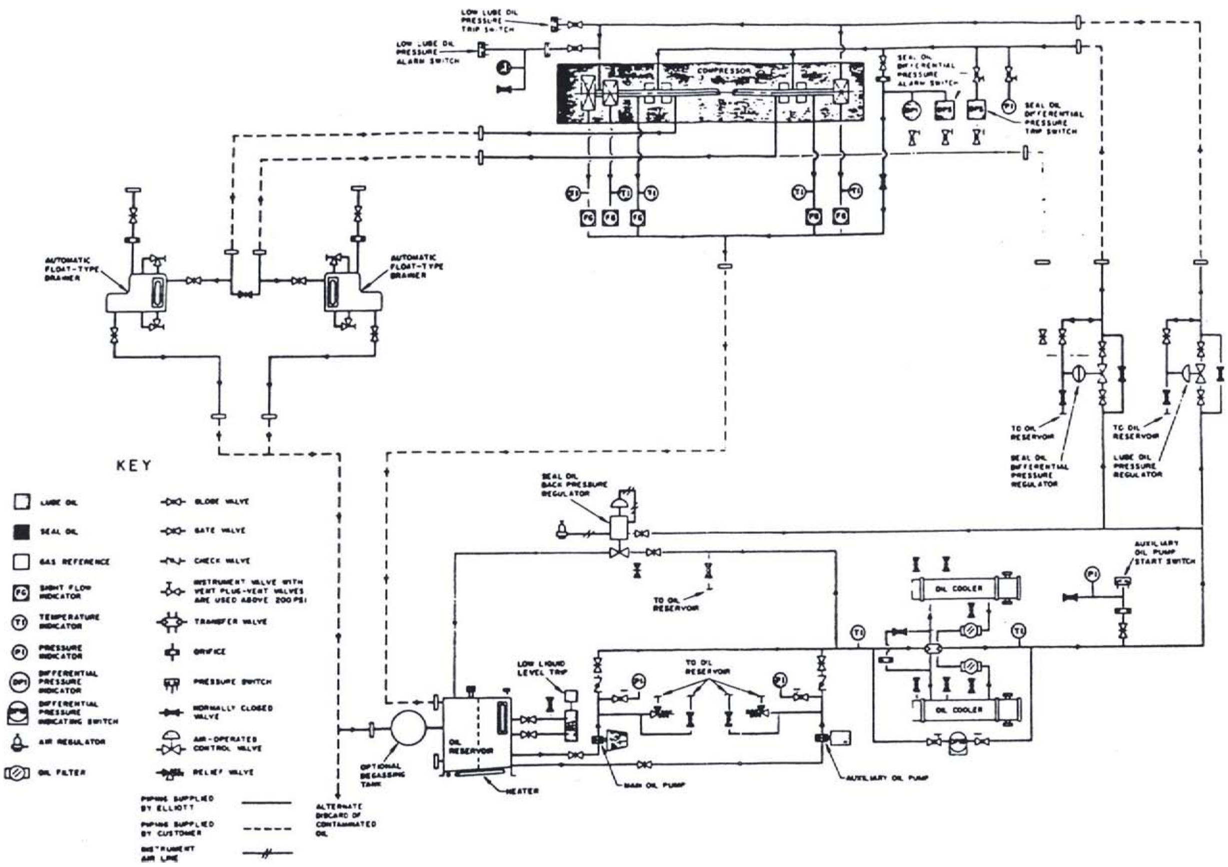

LOW LUBE OIL PRESSURE TRIP SWITCH
LOW LUBE OIL PRESSURE ALARM SWITCH
COMPRESSOR
SEAL OIL DIFFERENTIAL PRESSURE ALARM SWITCH
SEAL OIL DIFFERENTIAL PRESSURE TRIP SWITCH
AUTOMATIC FLOAT-TYPE DRAINER
AUTOMATIC FLOAT-TYPE DRAINER
TO OIL RESERVOIR
SEAL OIL DIFFERENTIAL PRESSURE REGULATOR
TO OIL RESERVOIR
LUBE OIL PRESSURE REGULATOR
SEAL OIL BACK PRESSURE REGULATOR
TO OIL RESERVOIR
AUXILIARY OIL PUMP START SWITCH
OIL COOLER
OIL COOLER
LOW LIQUID LEVEL TRIP
TO OIL RESERVOIR
OIL RESERVOIR
OPTIONAL DEGASSING TANK
HEATER
MAIN OIL PUMP
AUXILIARY OIL PUMP
ALTERNATE DISCARD OF CONTAMINATED OIL
KEY
LUBE OIL
SEAL OIL
GAS REFERENCE
SIGHT FLOW INDICATOR
TEMPERATURE INDICATOR
PRESSURE INDICATOR
DIFFERENTIAL PRESSURE INDICATOR
DIFFERENTIAL PRESSURE INDICATING SWITCH
AIR REGULATOR
OIL FILTER
GLOBE VALVE
GATE VALVE
CHECK VALVE
INSTRUMENT VALVE WITH VENT PLUG-VENT VALVES ARE USED ABOVE 200 PSI
TRANSFER VALVE
ORIFICE
PRESSURE SWITCH
NORMALLY CLOSED VALVE
AIR-OPERATED CONTROL VALVE
RELIEF VALVE
PIPING SUPPLIED BY ELLIOTT
PIPING SUPPLIED BY CUSTOMER
INSTRUMENT AIR LINE
FG
TI
PI
DPI
DPS

Figure 7.102 API G14 lube-seal oil system for ISO-carbon seals (Courtesy of Elliott Co.)

Start-up

In this example, the compressor case reference pressure during start-up is 100 psi. Therefore, the control valve will regulate pressure downstream of the filter and cooler to be equal to 150 psi. (Fifteen additional psi were added to the specified seal differential of 35 psi to account for the pressure drop between the exit of the seal system and the entrance to the seal chamber at the unit.) The concept to consider in this instance is that of an equivalent vessel (the discharge pipe of the seal oil supply system). The control valve must regulate the supply flow to the equivalent vessel such that the supply flow is equal to the flow demanded by the seal system to maintain a constant supply pressure. In the case of increasing demand flow and constant reference pressure, the pressure in the equivalent vessel would fall. The supply flow would be increased by a decreasing seal oil supply signal thus closing the bypass valve and providing additional supply flow. It can be seen that the functioning of the bypass control valve with a constant seal gas reference pressure, is identical to that of a lube system bypass valve.

Normal operation

For this example, normal operation requires a suction pressure of 500 lbs per square inch. As the suction pressure is increased, the differential pressure across the atmospheric seal, increases while the differential pressure across the gas side seal remains constant. In creasing differential pressure across the atmospheric seal will increase seal oil demand, thus resulting in reduced referenced oil pressure to the seal oil differential valve. This reduced pressure will act to close the valve and provide additional required seal oil to the seal system. If sufficient seal oil flow is not available when the bypass valve is completely closed, the referenced oil pressure will fall until the auxiliary pump starts. It is important to note that differential pressure switches must be used since the reference or seal downstream pressure can change during operation. If the seal oil differential continues to fall, the critical equipment will be tripped on low pressure. Typical set points for differential pressure control systems are to start the auxiliary pump to start at 35 psid and to trip the unit at 25 psid.

It is important to note that action to start the auxiliary seal oil pump and to trip the unit is based on differential seal oil to seal gas pressure. A sudden increase of seal gas reference pressure or a sudden decrease of seal oil pressure, will result in auxiliary pump start up and/or critical equipment trips. Care must be taken to assure the system is designed to prevent sudden oil pressure or reference gas pressure spikes that will lead to spurious trips. Seal systems tend to be softer than lube oil systems since the referenced pressure is a compressible fluid. That is, the requirement to start the auxiliary seal oil pump without tripping the unit tends to be easier and does not require accumulators.

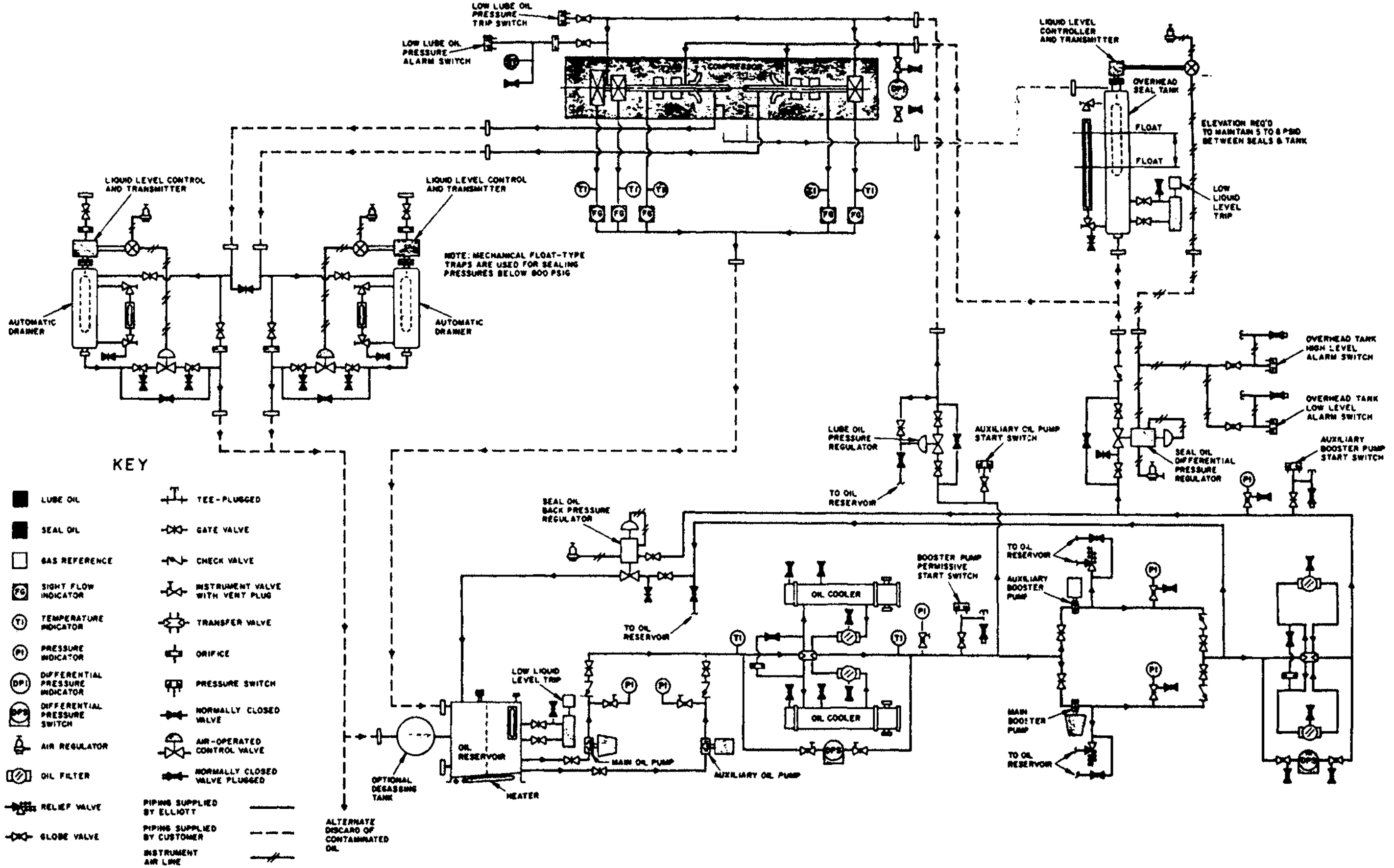

LOW LUBE OIL PRESSURE TRIP SWITCH
LOW LUBE OIL PRESSURE ALARM SWITCH
COMPRESSOR
DPI
LIQUID LEVEL CONTROLLER AND TRANSMITTER
OVERHEAD SEAL TANK
FLOAT
FLOAT
ELEVATION REQ'D TO MAINTAIN 5 TO 8 PSID BETWEEN SEALS & TANK
LOW LIQUID LEVEL TRIP
LIQUID LEVEL CONTROL AND TRANSMITTER
LIQUID LEVEL CONTROL AND TRANSMITTER
NOTE: MECHANICAL FLOAT-TYPE TRAPS ARE USED FOR SEALING PRESSURES BELOW 800 PSIG
AUTOMATIC DRAINER
AUTOMATIC DRAINER
TI
FG
OVERHEAD TANK HIGH LEVEL ALARM SWITCH
OVERHEAD TANK LOW LEVEL ALARM SWITCH
AUXILIARY BOOSTER PUMP START SWITCH
LUBE OIL PRESSURE REGULATOR
AUXILIARY OIL PUMP START SWITCH
TO OIL RESERVOIR
SEAL OIL DIFFERENTIAL PRESSURE REGULATOR
SEAL OIL BACK PRESSURE REGULATOR
TO OIL RESERVOIR
BOOSTER PUMP PERMISSIVE START SWITCH
TO OIL RESERVOIR
AUXILIARY BOOSTER PUMP
OIL COOLER
OIL COOLER
PI
DPS
MAIN BOOSTER PUMP
TO OIL RESERVOIR
LOW LIQUID LEVEL TRIP
OIL RESERVOIR
MAIN OIL PUMP
AUXILIARY OIL PUMP
HEATER
OPTIONAL DEGASSING TANK
ALTERNATE DISCARD OF CONTAMINATED OIL
KEY
LUBE OIL
SEAL OIL
GAS REFERENCE
SIGHT FLOW INDICATOR
TEMPERATURE INDICATOR
PRESSURE INDICATOR
DIFFERENTIAL PRESSURE INDICATOR
DIFFERENTIAL PRESSURE SWITCH
AIR REGULATOR
OIL FILTER
RELIEF VALVE
GLOBE VALVE
TEE-PLUGGED
GATE VALVE
CHECK VALVE
INSTRUMENT VALVE WITH VENT PLUG
TRANSFER VALVE
ORIFICE
PRESSURE SWITCH
NORMALLY CLOSED VALVE
AIR-OPERATED CONTROL VALVE
NORMALLY CLOSED VALVE PLUGGED
PIPING SUPPLIED BY ELLIOTT
PIPING SUPPLIED BY CUSTOMER
INSTRUMENT AIR LINE

Figure 7.103 API G14 lube-seal oil system for ISO-sleeve seals (Courtesy of Elliott Co.)

Level control valves

Function

Level control valves utilized in seal systems are designed to regulate seal oil flow to or from the seal in the same manner as s differential pressure control valve. Level control valves are used where extreme accuracy in seal oil to seal gas differential pressure is required. They are most commonly employed in bushing seal applications since bushing seal differential pressures are regulated to a minimum value in order to minimize internal seal oil leakage. Refer to Figure 7.103 which incorporates a level control valve used in the system incorporating bushing seals. Again, let's examine the start-up and normal operating case for this seal system configuration.

Start-up

Reference pressure for start-up is 100 psi. This reference pressure is established in the overhead seal oil tank and exerts a pressure of 100 psi on the liquid. Refer again to the concept of an equivalent vessel which is the level tank in this case. As the referenced pressure increases, the differential pressure across the atmospheric bushing of the seal will increase (the differential pressure against the gas side bushing will remain constant). This action will increase the atmospheric bushing seal flow which will reduce the level in the head tank. A reduction of head tank level sends a signal to the level control valve which will open the valve, thereby increasing flow to the supply side of the equivalent vessel unit the supply flow equals the seal demand flow and the equilibrium level is established.

As the suction pressure increases to 500 psi, the atmospheric bushing seal flow will increase proportionally, thereby causing a decease in head level. Again, a signal will be sent to the level control valve to open thus increasing supply flow to the equivalent vessel and re-establishing head level at the same point as before. In the event of referenced pressure decrease, the opposite situation will occur and a signal will be sent to the control valve to close, thereby decreasing supply flow. Remember, that liquid head is proportional to pressure. In this example, unequal demand and supply flows to the equivalent vessel cause changes in vessel liquid level which are pressure changes as previously discussed. Therefore, in this example the function of the level control valve is to act as a variable orifice to change system supply flow for different demand flow requirements to maintain a constant pressure (head) in the level tank.

Control valve location in the system

As seen in the preceding section, the location of the level control valves and differential pressure control valves can vary from design to design. Either valve can be used in any of the following locations:

- Bypass
- Supply
- Return

Bypass control

Depending on the type of seal system and the type of seal design used, the control valves will be positioned accordingly. If the seal design does not incorporate through flow, a differential bypass or bypass level control valve will be incorporated. The function of the valve in this position will be to bypass pump discharge flow and thus reduce seal supply flow. Note that this type of valve arrangement will be used with positive displacement type pumps only. If centrifugal pump were utilized, the valve would be incorporated in the pump discharge and would regulate differential pressure downstream of the valve.

Supply control

If the seal arrangement incorporates flow through with a back pressure orifice, differential pressure supply control or supply level control is incorporated. If the seal is a busing type, level control would be utilized. If the seal is a contact type, differential pressure control would be used. The function of this system is to control supply flow to the seal system in accordance with seal system demand.

As an example, if seal reference pressure increases, atmospheric seal flow will increase and seal system demand will increase. This action will open the supply control valve, thus allowing supply flow to equal demand flow and achieving equilibrium supply system differential pressure.

Return control

If the seal configuration incorporates a flow through system without a fixed orifice, the control valve will be located in the discharge to compensate for oil to gas differential pressure changes.

As an example if a flow through bushing seal combination were used and the seal reference gas reference pressure increased, the atmospheric seal flow would increase, thereby increasing seal demand. In this configuration, seal supply flow is fixed, therefore, increased seal demand requires decreased flow through demand. This action would immediately result in a lower seal oil supply pressure and would close the control valve resulting in a reduced flow through quantity to compensate for the increased atmospheric seal requirement. Refer to Figure 7.104 for a schematic of this system.

Regardless of the location of the control valves in a seal system, the instrumentation is identical, that is, starting auxiliary pumps on decreasing level or differential pressure and tripping the critical equipment on decreasing level or pressure.

This concludes the discussion of the supply side component sizing of a seal system. In the next chapter, we will direct our attention to the return side of the seal system.

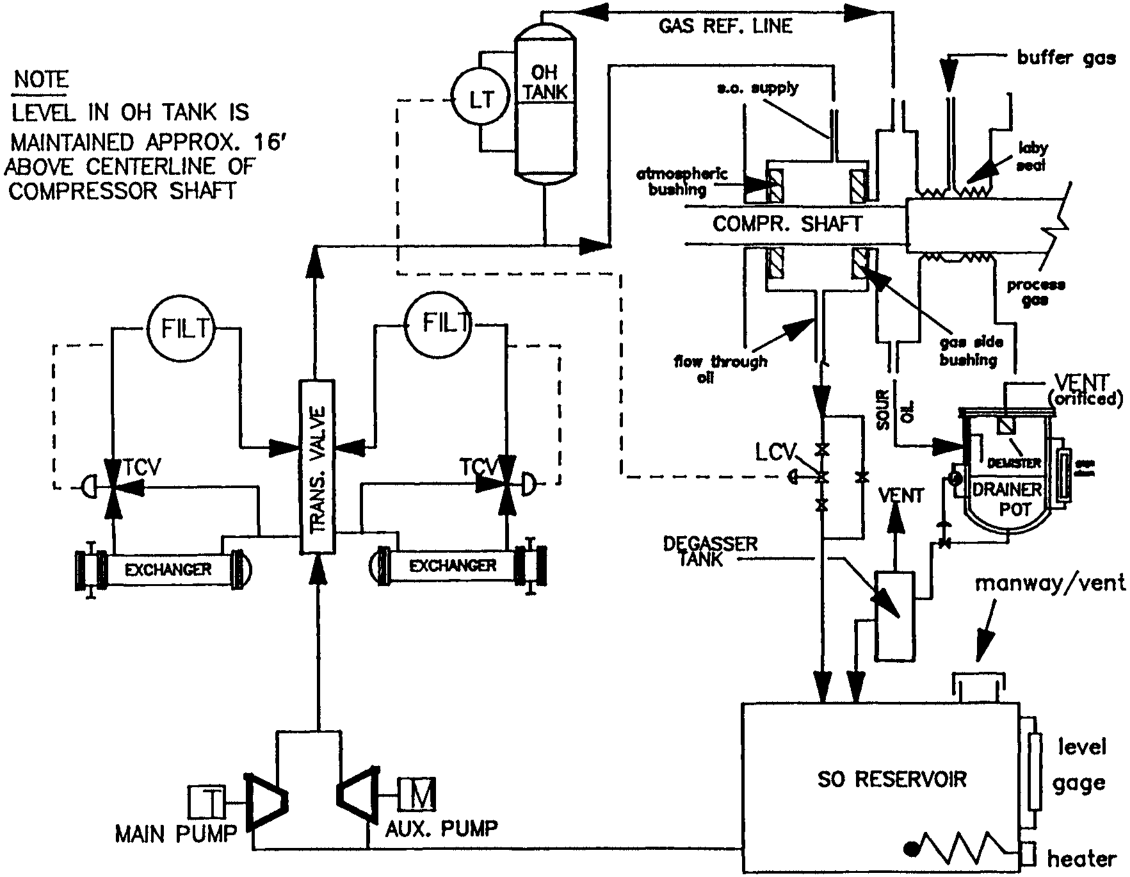

NOTE
LEVEL IN OH TANK IS MAINTAINED APPROX. 16' ABOVE CENTERLINE OF COMPRESSOR SHAFT
GAS REF. LINE
OH TANK
LT
buffer gas
s.o. supply
laby seal
atmospheric bushing
COMPR. SHAFT
process gas
gas side bushing
flow through oil
SOUR OIL
VENT (orificed)
DEMISTER
DRAINER POT
LCV
VENT
DEGASSER TANK
manway/vent
FILT
FILT
TCV
TCV
TRANS. VALVE
EXCHANGER
EXCHANGER
SO RESERVOIR
level gage
heater
T
M
MAIN PUMP
AUX. PUMP

Figure 7.104 Typical seal oil system (Courtesy of M.E. Crane Consultant)

Seal system – the contaminated seal oil drain system

(Compressor liquid seal component knowledge)

- Introduction
- Basic system configuration
- Oil drain system component design
- System reliability considerations

Introduction

The return side of a seal oil system consists of three distinct return lines.

- The atmospheric drain back to the seal oil reservoir.
- The pressurized drain back to the seal oil control valves.
- The contaminated seal oil drain.

All liquid seals are designed such that a small amount of oil leakage (10–20 gallons per day per seal) enter the drain between the inboard gas dies seal and the compressor internals. This drain is commonly known as the contaminated oil drain. To assure that gas does not leak into the seal, a small amount of liquid is designed to continuously flow through the seal. Depending on the nature of the process gas, the oil may or may not be recirculated to the seal oil tank. The function of the contaminated oil drain system is therefore to collect all contaminated oil and direct this oil to the desired location. This may be directly to the seal oil reservoir, to a degassing tank or other oil reclamation device, or to a contaminated oil drain. We will now examine basic contaminated oil drain system configurations and discuss the function of each component.

Basic system configuration

Refer to Figure 7.105 which shows a basic contaminated oil drain system. The system consists of:

- The contaminated oil drain line to a seal oil drainer for each seal in the unit.
- The drainer.

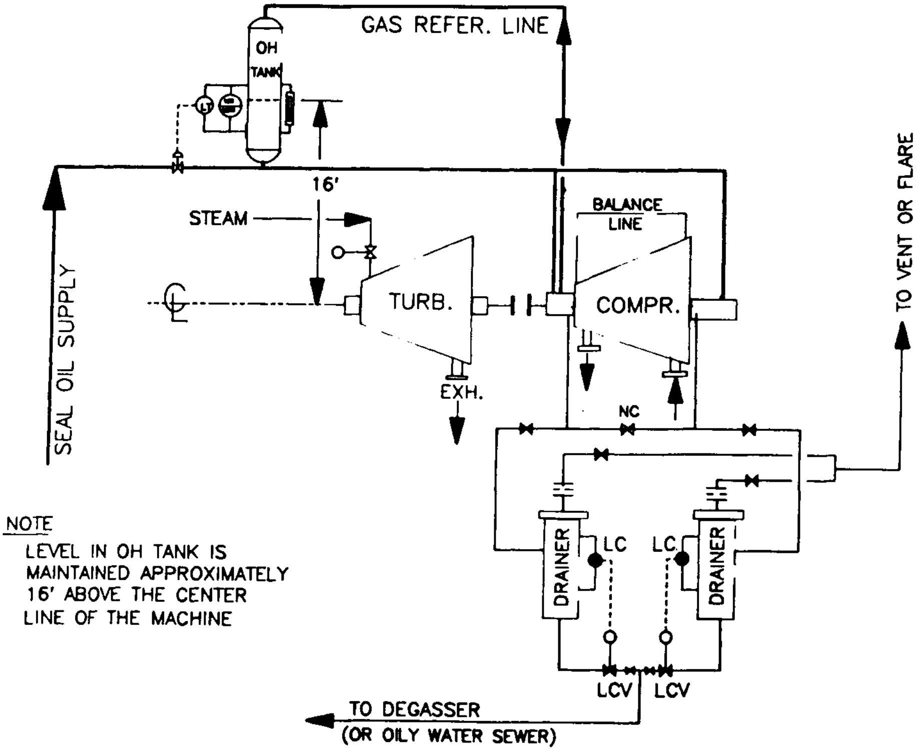

GAS REFER. LINE
OH TANK
16'
STEAM
BALANCE LINE
TURB.
COMPR.
EXH.
NC
SEAL OIL SUPPLY
TO VENT OR FLARE
DRAINER
LC
LC
DRAINER
LCV
LCV
TO DEGASSER
(OR OILY WATER SEWER)
NOTE
LEVEL IN OH TANK IS MAINTAINED APPROXIMATELY 16' ABOVE THE CENTER LINE OF THE MACHINE

Figure 7.105 Schematic – seal oil arrangement (Courtesy of M.E. Crane Consultant)

- The vent connection or reference connection from the top of the drainer.
- The drainer level control device (internal or external) which automatically drains upon reaching a preset level.
- The degassing tank (optional).
- The oil reclamation device (optional).

The following system design alternatives are available depending upon the operating conditions of the compressor and the process gas.

Sweet hydrocarbon or inert gas service

For sweet or inert gas service, the seal oil drain can be returned directly to the reservoir provided the drainers are sized for adequate residence time and the seal oil leakage is reasonable (less than one gallon per hour per seal). A sweet hydrocarbon gas is defined as a gas that does not contain hydrogen sulfide (H_2S). The vent line on top of the drainer can be routed to a lower pressure source, atmosphere or back to the compressor suction. If routed back to the compressor suction, a demister should be installed to prevent oil from entering the compressor case. The sizing of the orifice in the vent line of each drainer is critical in that it assures that all contaminated oil flow will enter the drainer. Too low a velocity will allow contaminated oil to enter the compressor. Too high a velocity could cause oil to enter the compressor via the vent or reference line.

High suction pressure

In high suction pressure applications, the system configuration changes somewhat as shown in Figure 7.106.

In this application, an external level control valve is recommended on each contaminated seal oil drainer. In addition, venting drainers to atmosphere or to flare will result in exceptionally high pressure drop across the orifice. In this application, the vent line is usually routed back to the compressor suction. Again, an adequately sized demister must be installed to prevent oil ingestion in the compressor.

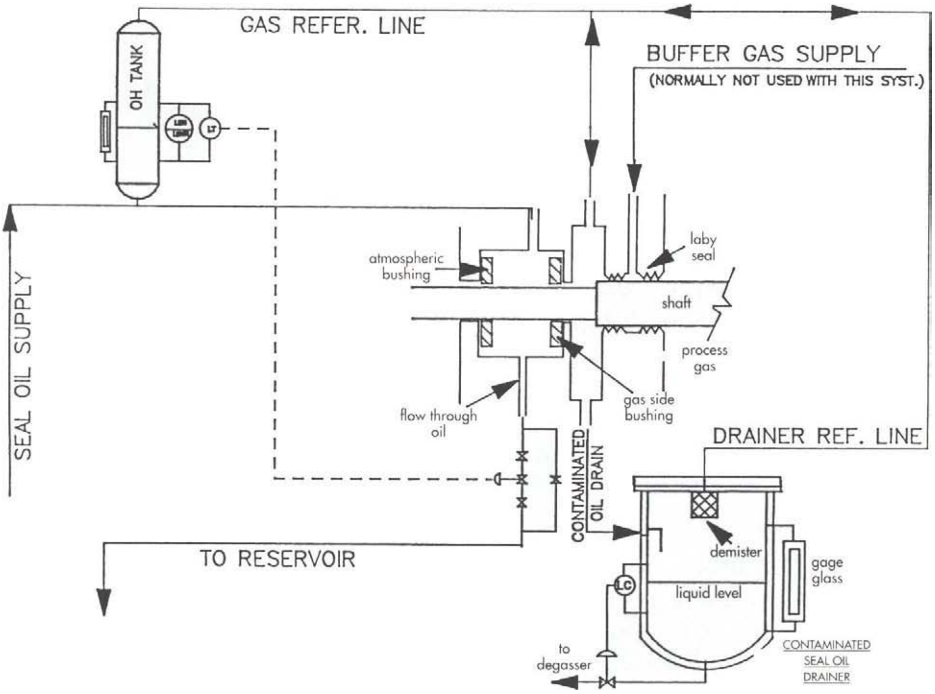

GAS REFER. LINE
OH TANK
LT
BUFFER GAS SUPPLY
(NORMALLY NOT USED WITH THIS SYST.)
SEAL OIL SUPPLY
atmospheric bushing
laby seal
shaft
process gas
gas side bushing
flow through oil
DRAINER REF. LINE
CONTAMINATED OIL DRAIN
TO RESERVOIR
demister
gage glass
LC
liquid level
to degasser
CONTAMINATED SEAL OIL DRAINER

Figure 7.106 Contaminated oil drain system (with referenced drainer) (Courtesy of M.E. Crane Consultant)

Vacuum service

Compressors that act with suction pressure below atmospheric pressure or use a suction throttle valve, can have a reference pressure less than atmospheric. In this case, the contaminated seal oil drainer vent line must be referenced back to the compressor suction, in order to assure proper operation and not allow air to enter the demister through the vent line in the reverse direction. For this application, a buffer gas system must be installed that will be designed to maintain the drainer pressure above atmospheric pressure at all times in order to allow the drainer to be drained.

Sour process gas

Where the process gas can contaminate the seal oil leakage (as in the case of H_2S gas, etc.), the contaminated seal oil drain line is usually routed to the plant contaminated oil system and not back to a degassing tank or the reservoir.

Oil drain system component design

The function, sizing criteria and operating specifics of each major component of the contaminated oil drain system will now be discussed.

The contaminated seal oil drainer

The function of the contaminated seal oil drainer as shown in Figure 7.107 is to contain all of the oil leakage from a specific seal. Normally, drainers are automatic, that is, they drain oil when a specific pre-set level in the drainer is reached. The typical sour oil leakage per day varies from less than five gallons to an excess of 20 gallons per day. Normal vessel capacity of liquid is approximately 1/4– 1/2 gallon. Consequently, drainers must either be manually drained or automatically drained between 40–80 times a per day. The inlet connection to the contaminated seal oil drainer is from the gas side seal oil drain and contains an oil-gas mixture. The gas is either the process gas or a clean buffer gas as will be discussed in following sections. The top of the drainer contains a vent connection which may be routed to a reference connection in the compressor case or directly to vent. Regardless of the configurations, the drainer contains a mixture of free gas above the liquid and entrained gas in the liquid. Drainers are essentially of two designs.

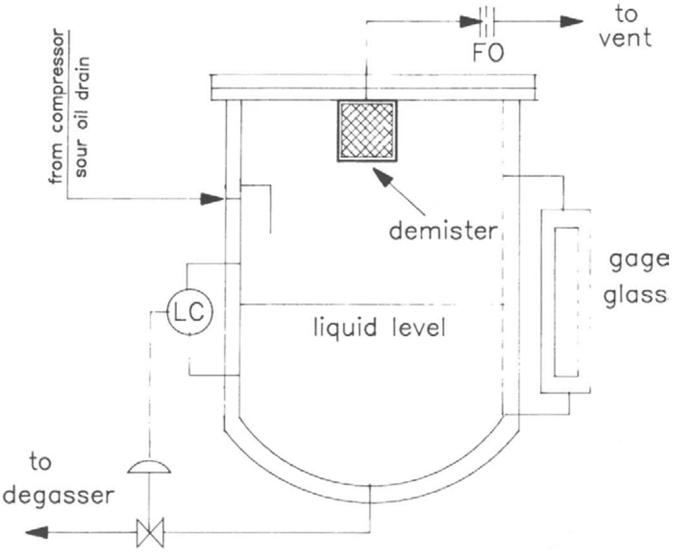


Figure 7.107 Contaminated seal oil drainer (Courtesy of M.E. Crane Consultant)

Internal valve

This design incorporates a ball float valve shown in Figure 7.108 which is internal to the drainer and opens at a prescribed level. In order to control the rate of drainage from the drain pot with the valve open, an orifice is sized. Note that this orifice is sized to allow controlled drainage time under normal operating conditions. Conceptually, this orifice represents another atmospheric bushing in the seal oil system. It is similar to the atmospheric seal in that the differential across the orifice will vary with reference pressure in the compressor. Many times care is not given to adequately sizing this orifice. As a result, during start-up conditions, with low compressor case suction pressure, there is insufficient pressure drop across this orifice to adequately drain the vessel. The vessel will not drain until there is sufficient pressure drop. That is, there must be a column of liquid high enough (head) to drain through this orifice. Considering the installation location of seal oil drainers, many times the height available from the center line of the compressor down to the drainer is insufficient at low or zero pressure conditions to force a drain. Therefore, under these conditions, all contaminated seal oil will drain into the compressor case. As can be seen from Figure 7.109, drainer installations should be provided with bypass valves around the drainer. It may be necessary to open bypass valves during low pressure operation to allow drainage back to the reservoir. *Caution must be exercised during this operation if compressor gas is toxic or flammable since a full stream of gas will be exiting continuously back to the appropriate vessel.*

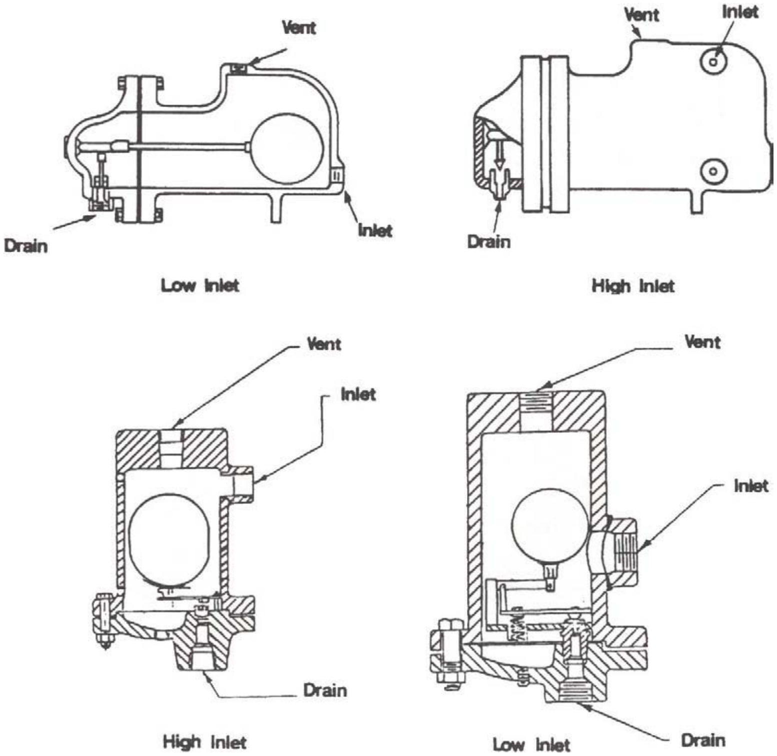


Figure 7.108 Ball float valve. Above: WKM drainers; below: Armstrong drainers (Courtesy of Elliott Co.)

Caution: Prior to opening any system connections or sampling fluids, obtain a site work permit to assure the area and conditions are safe for work or entry.

A properly sized orifice for low pressure conditions must be installed in this line to minimize loss of gas.

Contaminated seal oil drainer with external valves

This application is usually utilized in higher pressure cases such a re-injection compressors where suction pressure and seal oil drainer pressure run between 1,000 and 1,500 lbs per square inch. The external valve is controlled by a level control transmitter which sends a signal to open the valve when the level reaches a specified amount and quickly closes the valve when the level falls below a specified set point. Again, care must be given in sizing the valve to assure that sufficient valve area is available under low suction pressure conditions to allow drainage back to the appropriate vessel. In addition to the valve,

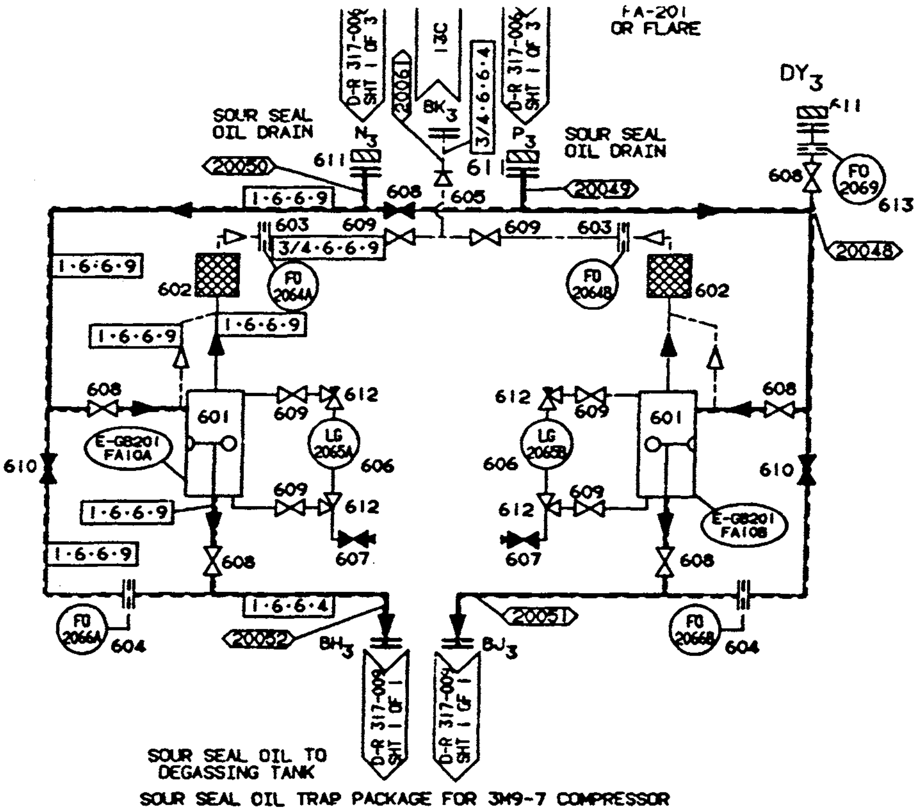

FA-201
OR FLARE
D-R 317-006
SHT 1 OF 3
20061
13C
BK3
3/4·6·6·4
D-R 317-006
SHT 1 OF 3
SOUR SEAL
OIL DRAIN
N3
P3
SOUR SEAL
OIL DRAIN
DY3
611
20050
20049
1·6·6·9
608
605
603
609
3/4·6·6·9
FO
2064A
FO
2064B
602
613
FO
2069
20048
601
612
LG
2065A
LG
2065B
606
607
610
E-GB201
FA10A
E-GB201
FA10B
604
FO
2066A
FO
2066B
1·6·6·4
20052
20051
BH3
BJ3
D-R 317-009
SHT 1 OF 1
SOUR SEAL OIL TO
DEGASSING TANK
SOUR SEAL OIL TRAP PACKAGE FOR 3M9-7 COMPRESSOR

Figure 7.109 Sour seal oil trap arrangement (Courtesy of Dresser-Rand)

downstream piping must be adequately sized and designed to allow contaminated seal oil to exit the drainer.

Drainer reliability considerations – As can be seen, the reliable operation of the drainer valve, internal or external, is essential to safety and reliability. The valve must open and close tightly upon signal to assure a minimum amount of processed gas exits the drainer. Many applications process a gas that tends to be sticky or has a high amount of carbon that will cause internal valves to bind, thus keeping the valve open at all times.

Attention must be paid to the gage glass on the drainer. Gage glasses must be kept clean so that level can be observed. If level is not present, the drainer exit should be checked to assure that gas is not exiting the drainer. If this is the case, the drainer should be isolated and inspected. *Caution – any action involving opening valves and drainers requires an area safety permit.*

Systems should be designed such that drainers can be isolated during operation and one drainer can temporarily service two seals so that maintenance can be performed on a drainer while the unit is in operation. If a continued problem is experienced with clogging of drainer orifices or hanging up of internal valves, consideration should be given to injecting a clean buffer gas which will assure satisfactory operation of the drainer. Note that external valves are also subject to this malfunction in that debris can enter the valve causing it to remain open.

Another reliability consideration is the drainer bypass line. Many times this line is inadvertently left open, either after start-up or opened by operators during operation. This valve should be closed during operation. Otherwise a continuous stream of gas exits and proceeds downstream.

If the drainer gage glass is continually full, this could indicate that the drain valve is not properly opening and contaminated oil could be forced into the compressor under this condition. Many processes prohibit the entrance of oil into the compressor and this action could cause shutdown of the unit. Again, the drainer should be isolated and inspected. Failure to properly monitor and maintain drainers leads to many unscheduled shutdowns of critical equipment.

One instance of false level gauge indication is shown in Figure 7.110. In this case, the top reference line of the level gage is referenced to the drainer vent line. In this application, the vent line was referenced to the suction vessel and the velocity through the orifice was great enough to create a vacuum on the gas in between the top reference line take off and the fluid in the level gauge. The result was that the liquid level was actually forced to the top of the glass (by greater pressure in the drainer), creating a false indication of a full oil drainer. The solution was to install a larger (four inch diameter) section at the drainer vent connection to minimize the velocity in this area. The high velocity steam acted as an eductor creating a lower pressure on the top of the liquid in the level gauge.

As an exercise, let's calculate the reduction in gas pressure on top of the oil in this application that would allow the oil level to rise six inches (the height of the level glass). Solving for pressure,

$$P = \frac{HD \times S.G.}{2.311}$$

$$= \frac{.5(ft) \times .85}{2.311}$$

$$= 0.184 \text{ psi}$$

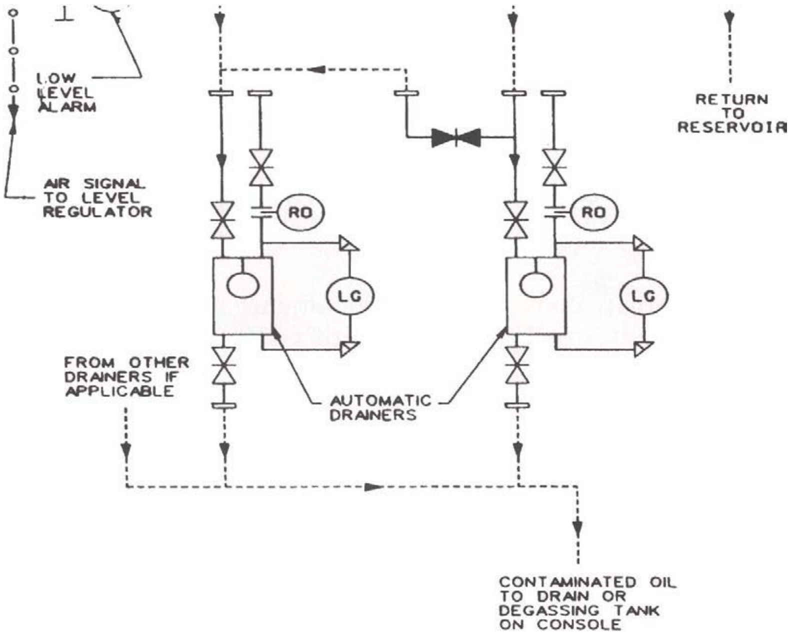

LOW LEVEL ALARM
AIR SIGNAL TO LEVEL REGULATOR
RETURN TO RESERVOIR
RO
RO
LC
LC
FROM OTHER DRAINERS IF APPLICABLE
AUTOMATIC DRAINERS
CONTAMINATED OIL TO DRAIN OR DEGASSING TANK ON CONSOLE

Figure 7.110 False level gauge indication (Courtesy of Elliott Co.)

Therefore, if the high velocity gas stream can reduce the pressure in the trapped volume between the gas stream and the top of the liquid level by .184 psi, the level will rise to the top of the gage glass and give the illusion of a full drainer. To see if this problem exists, on line, briefly close the inlet valve to the drainer. If the level suddenly decreases, this would indicate this type of problem. *Caution: immediately open drainer inlet line to avoid seal oil entering the compressor.*

Demisters

When the vent line from the top of the drainers is referenced back to any section of a compressor, a demister must be installed to minimize the amount of oil entering the compressor. The functions of demisters, therefore, are to eliminate oil migration into the compressor via the vent line. The demisters must be properly sized to assure their efficiency. Care must be given to calculating the velocity through the demister which is dependent on the vent orifice. Maximum differential pressure conditions across the orifice must be considered. Frequently, mesh demisters are designed to be integral with the contaminated seal oil drainer as shown in Figure 7.107. In this case, any oil mist exiting the top of the drainer will be condensed and will fall back into the drainer. It must be understood that demisters are not 100% efficient. If the process cannot tolerate any seal oil, the vent line should be routed directly to flare or to atmosphere, depending upon the gas composition. *Caution – any toxic or flammable gas must be routed to a safe location.*

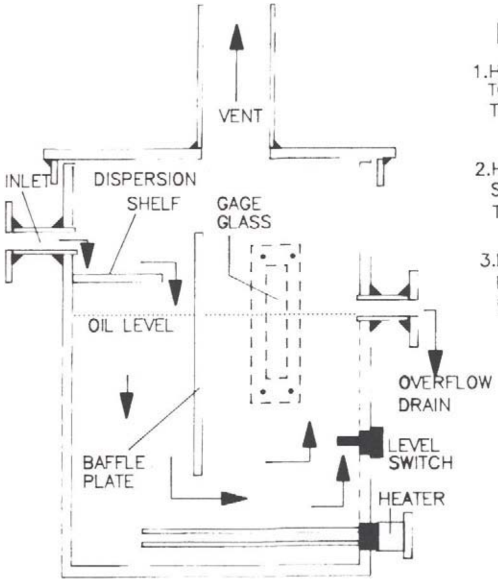
Degassing tanks

As previously mentioned, the oil that enters the contaminated seal oil drainer is accompanied by free gas which exits the drainer through the vent, and entrained gas in the oil. The oil must be properly degassed prior to entrance back into the seal oil reservoir. The function of a degassing tank, therefore, is to degas the contaminated seal oil so that all oil exiting the degasser is within the original oil specification. A typical degassing tank is shown in Figure 7.111. The tank contains baffles, a heating device, and an overflow drain with a properly sized vent in order to degas all seal oil entering this vessel. Experience has shown that a degasser sized for 72 hours residence time, based on the total estimated sour oil leakage, usually is sufficient to achieve the design objective.

As an example, if a compressor containing two seals each has a maximum specified leakage of 20 gallons per seal per day, the degassing tank should be sized as follows:

capacity = total leakage per day × 3 days = 120 gallons.

A cursory inspection of any refinery or chemical plant will show that most degassing tanks are undersized. As a result, seal oil sampling



NOTES

1.HEATER IS THERMOSTATICALLY CONTROLLED TO MAINTAIN OIL IN DEGASSER AT 180 DEG TO ENHANCE DEGASSING.

2.HEATER IS INTERLOCKED WITH LEVEL SWITCH SO HEAT IS NOT APPLIED UNLESS THERE IS A LEVEL IN TANK.

3.DEGASSER IS SIZED TO PROVIDE 72 HOURS RESIDENCE TIME BASED ON THE TOTAL ESTIMATED SOUR OIL LEAKAGE.

Figure 7.111 Typical degasser arrangement (Courtesy of M.E. Crane Consultant)

usually shows a deterioration of oil viscosity and flash point. As mentioned in the previous chapter, flash point is the temperature at which the oil will sustain combustion. Light gasses, (hydrogen, and hydrogen mixtures), significantly reduce oil flash points. Experience has shown that this value can approach the operating temperature of the system! It is strongly recommended that the following action should be taken when seal oil reservoir samples indicate a low flash point.

- Temporarily isolate seal oil drainer return, collect all seal oil and vacuum degas the seal oil. Provide make up fresh seal as required.
- Adequately size a degassing tank and install at earliest opportunity.
- Consider the installation of an oil reclamation device.

Figure 7.112 is a table of mineral oils used for seal oil service.

Characteristic	Light	Medium
Gravity, api	31.7	30.6
Pour, °F (°C)	20 (-7)	20 (-7)
Flash, °F (°C)	395 (201)	400 (204)
Viscosity		
Sus at 100°F	150/165	215/240
Sus at 210°F	44	48.8
Csi at 40°C	28.8/32.0	41.4/46.0
Csi at 100°C	5.2	6.5
VI, min	95	95
Iso viscosity grade	32	46
Color, astm. max	1.5	2.0
Neutralization number, max.	0.20	0.25
Rust test (A&B) (astm D665-IP135)	Pass	Pass
Demulsibility		
(astm 1401) 3 ml max.		
at 130°F (54°C) $^{1}/_{2}$ hr.	Pass	Pass
at 180°F (82°C) 1 hr		

Figure 7.112 Typical oil flash points for seal oils

Please note the values of the oil flash points and remember that many operating seal oil systems contain oil flash points that are on the order of 120°F. This is particularly dangerous in the case of combined lube and seal oil systems where the oil in the reservoir will actually enter the bearing system.

Oil reclamation units

In cases where the degassing tanks have proven not to be effective, or are inadequately sized, the use of an oil reclamation unit should be considered. All oil from drainers should be collected or directly piped to an oil reclamation unit. Considering the cost of typical mineral oil (approximately $25 per gallon), a standard compressor with two seals can use a $1,000 of oil per day if oil cannot be returned to the reservoir.

Figure 7.113 shows a typical oil reclamation unit which has capability to degas all oil entering the unit. In large installations, this unit may be justified for direct installation downstream of the drainers. For smaller systems, the purchase of one unit should be considered for the site. Gas entrained liquids can then be collected and transported to the unit for reclamation at specified intervals.

System reliability considerations

The contaminated oil drain system is one of the most neglected subsystems in a compressor seal oil system. It is out of the way, its function is not fully understood, and it is not usually monitored with accuracy. Following are a few suggestions concerning drainer system reliability:

1. Care should always be given to the inspection of site glasses.
2. Oil samples to determine adequate degassing should be taken periodically.
3. Leakage rates from the drainer should be regularly measured to determine the condition of the seals.

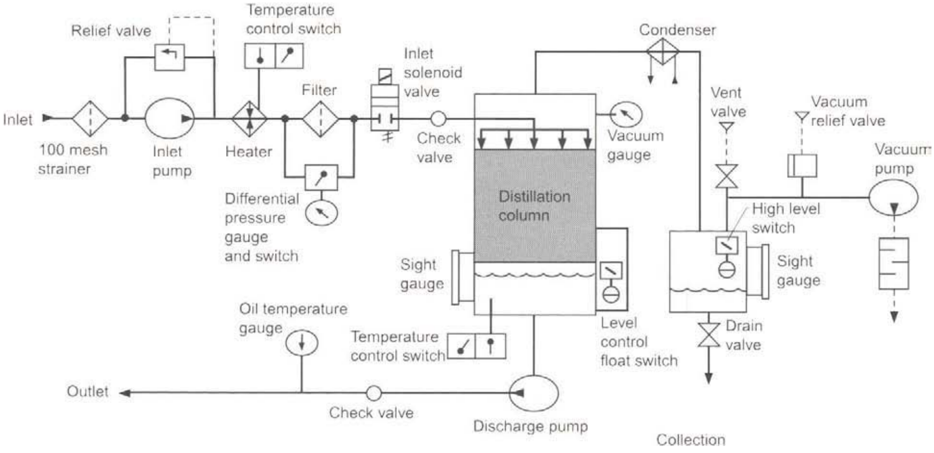

Relief valve
Temperature control switch
Condenser
Inlet solenoid valve
Filter
Inlet
100 mesh strainer
Inlet pump
Heater
Check valve
Vacuum gauge
Vent valve
Vacuum relief valve
Vacuum pump
Differential pressure gauge and switch
Distillation column
High level switch
Sight gauge
Sight gauge
Oil temperature gauge
Temperature control switch
Level control float switch
Drain valve
Outlet
Check valve
Discharge pump
Collection

Figure 7.113 Oil reclamation unit schematic (Courtesy of Petroneics, Inc.)

4. The atmospheric side seal drain should incorporate site glasses as a means of monitoring flow visually to determine if flow quantities have significantly changed.
5. It must be remembered that control valves can be used as rough flow meters when pressure across the valve and valve travel are known. A control valve can give an indication of change of flow rates in the system and will indicate any change in seal oil flows. It must be remembered that the atmospheric side seal oil flows will vary with changing with reference pressure. However, in most installations once units are on-line, reference pressure is relatively constant. Therefore, significant changes in seal oil control valves position will indicate deterioration of seals.

This concludes our discussion of the contaminated oil drainage system. We will further discuss this system when the subject of buffer gas systems is studied in subsequent chapters.

Dry gas seal systems

(Compressor dry gas seal component knowledge)

- Introduction
- Dry gas seal design
- Gas seal system types
- Summary

Introduction

Thus far, we have been studying in detail all of the components required for an auxiliary system. Many times, the reliability of critical equipment is dependent on the reliability of each component in every auxiliary system connected with the critical equipment unit. How do we maximize critical equipment reliability? The easiest way is to eliminate the auxiliary systems. Imagine the opportunity to eliminate all of the components; pumps, filters, reservoirs, etc. and thereby increase reliability and hopefully, the safety of the equipment. The gas seal as used in compressor applications affords the opportunity to achieve these objectives. However, the gas seal is still part of a system and the entire gas seal system must be properly specified, designed, maintained and operated to achieve the objectives of optimum safety and reliability of the critical equipment.

In this section, the principles of gas seal design will be discussed and applied to various gas seal system types.

System function

The function of a gas seal system is naturally the same as a liquid seal system. In fact, we have defined such systems as fluid seal systems. To repeat, the function of a fluid seal system, remembering that a fluid can be a liquid or a gas, is to continuously supply clean fluid to each specified seal interface point at the required differential pressure, temperature, and flow rate. Therefore, one would expect the design of a gas seal and a liquid seal to be very similar, which, in fact, they are. Then why are their systems so different?

Comparison of a liquid and gas sealing system

Figure 7.114 shows a liquid sealing system as previously discussed in this section.

Compare this system to Figure 7.115 which shows a gas seal system, if

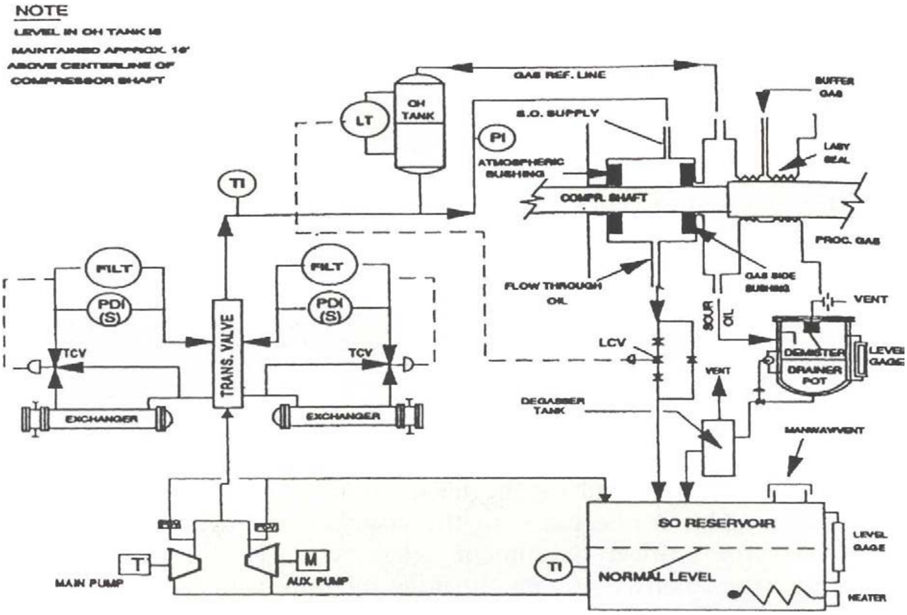

NOTE
LEVEL IN OH TANK IS
MAINTAINED APPROX. 16'
ABOVE CENTERLINE OF
COMPRESSOR SHAFT
GAS REF. LINE
BUFFER GAS
OH TANK
LT
S.O. SUPPLY
PI
LABY SEAL
ATMOSPHERIC BUSHING
TI
COMPR. SHAFT
PROC. GAS
FILT
FILT
GAS SIDE BUSHING
FLOW THROUGH OIL
VENT
PDI (S)
PDI (S)
SOUR OIL
TRANS. VALVE
LCV
TCV
TCV
DEMISTER
LEVEL GAGE
DRAINER POT
VENT
EXCHANGER
EXCHANGER
DEGASSER TANK
MANWAY/VENT
SO RESERVOIR
LEVEL GAGE
T
M
TI
MAIN PUMP
AUX. PUMP
NORMAL LEVEL
HEATER

Figure 7.114 Typical seal oil system (for clearance bushing seal)

the same compressor were retrofitted for a gas seal. WOW!! What a difference. Why are there such a small amount of components for the gas seal system? As an aid, refer to Figure 7.116 which shows a typical pump liquid flush system as specified by the American Petroleum Institute. This system incorporates a mechanical seal and utilizes pump discharge liquid as a flush for the seal. Refer now to Figure 7.115 and observe the similarities. It should be evident that a gas seal system is simplified in compressor applications over a liquid seal system merely because the gas seal utilizes the process fluid. This is exactly the same case for a pump. By using the process fluid, and not a liquid, one can eliminate the need to separate liquid from a gas, thereby totally eliminating the need for a liquid supply system and the need for a contaminated liquid (sour oil), drain system.

Referring back to Figure 7.114, therefore, we can see that the following major components are eliminated:

1. The seal oil reservoir
2. The pumping units
3. The exchangers
4. The temperature control valves
5. The overhead tank

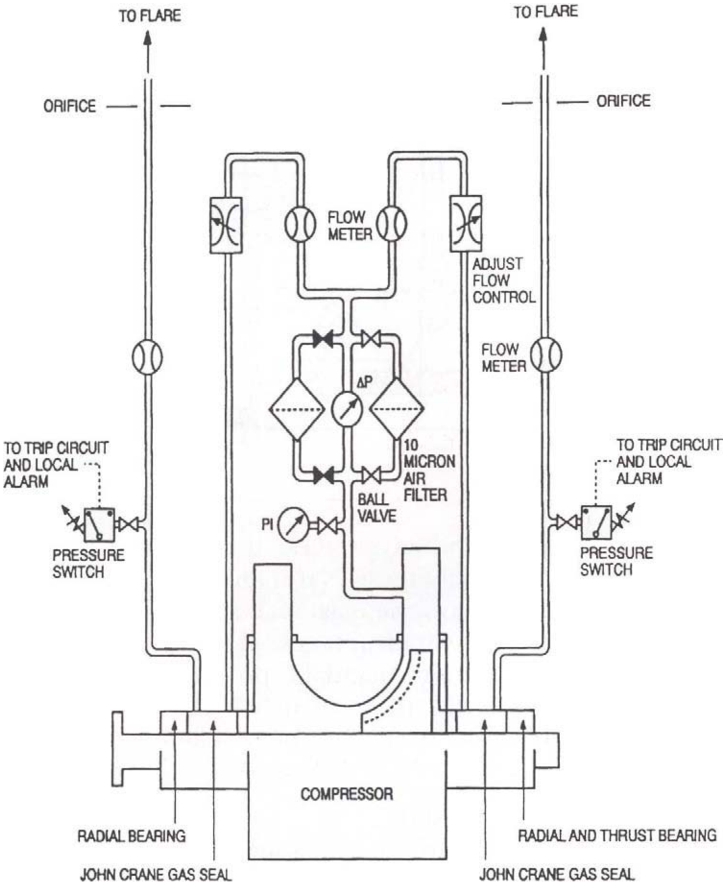

TO FLARE
TO FLARE
ORIFICE
ORIFICE
FLOW METER
ADJUST FLOW CONTROL
FLOW METER
ΔP
10 MICRON AIR FILTER
BALL VALVE
PI
TO TRIP CIRCUIT AND LOCAL ALARM
TO TRIP CIRCUIT AND LOCAL ALARM
PRESSURE SWITCH
PRESSURE SWITCH
COMPRESSOR
RADIAL BEARING
RADIAL AND THRUST BEARING
JOHN CRANE GAS SEAL
JOHN CRANE GAS SEAL

Figure 7.115 Typical gas seal system (Courtesy of John Crane Co.)

6. The drain pot
7. The degassing tank
8. All control valves
9. A significant amount of instrumentation

Referring back to the function definition of the gas seal system, all requirements are met. ‘Continuously supplying fluid’ is met by utilizing the discharge pressure of the compressor. The requirements for ‘specified differential pressure, temperature and flow rate’ are met by the design of the seal itself which can accommodate high differential pressures, high temperatures, and is sized to maintain a flow rate that will remove frictional heat necessary to maintain seal reliability. The only requirement not met is that of supplying a clean fluid, and this can be seen in Figure 7.115. This requirement is met by using a dual filter.

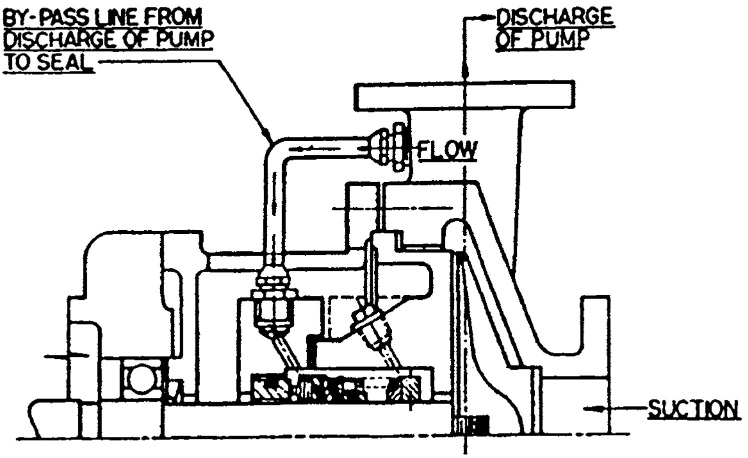


Figure 7.116 Liquid seal flush (Courtesy of John Crane Co.)

When one considers all the advantages, the next question to ask is, okay, what are the disadvantages? Naturally, there are disadvantages. However, proper design of the gas seal system can minimize and eliminate many of the disadvantages. Do not forget that the requirements for any system mandate proper specification, design, manufacture, operation and maintenance. One can never eliminate these requirements in any critical equipment system.

Considerations for system design

As mentioned above, there are disadvantages to a gas seal system which are not insurmountable but must be considered in the design of such a system. These considerations are as follows:

Sensitivity to dirt – since clearances between seal faces are usually less than 0.0005 inch and seal design is essential to proper operation, the fluid passing between the faces must be clean (5–10 microns maximum particle size).

Lift-off speed – as will be explained below, a minimum speed is required for operation. Care must be taken in variable speed operation to assure that operation is always above this speed.

Positive prevention of toxic gas leaks to atmosphere – since all seals leak, the system must be designed to preclude the possibility of toxic of flammable gas leaks out of the system. This will be discussed in detail below.

Possible oil ingestion from the lube system – a suitable separation seal must be provided to eliminate the possibility of oil ingestion from the bearings. Whenever a gas seal system is utilized, the design of the critical equipment by definition incorporates a separate lube oil and seal system. Consideration must be given during the design or retrofit phases to the separation between the liquid (lube) and gas seal system.

If all of the above considerations are incorporated in the design of a gas seal system, its reliability has the potential to exceed that of a liquid seal system and the operating costs can be reduced.

Before moving to the next section, however, one must consider that relative reliability between gas and liquid seal systems are a function of proper specification, design, etc. as mentioned previously. A properly designed liquid seal system that is operated and maintained as detailed in this book can achieve reliabilities of a gas seal system. Also, when one considers operating costs of the two systems, various factors must be considered. While the loss of costly seal oil is positively eliminated, with a gas seal system (assuming oil ingestion from the lube system does not occur) the loss of process gas, while minimal, can be expensive. It is argued that the loss of process gas from a liquid seal system through drainer vents and degassing tank vents, is also significant. While this may be true in many cases, a properly specified, designed and operated liquid seal system can minimize process gas leakage such that it is equal or even less than that of a gas seal.

There is no question that gas seal systems contain far fewer components and are easier to maintain than liquid seal systems. These systems will be used extensively in the years ahead. The intention of this discussion is to point out that existing liquid seal systems that cannot be justified for retrofit or cannot be retrofitted easily, can be modified to minimize outward gas leakage and optimize safety and reliability.

Dry gas seal design

Principles of operation

The intention of this sub-section is to present a brief detail of the principles of operation of a dry gas seal in a conceptual form. The reader is directed to any of the good literature available on this subject for a detailed review of gas seal design.

Refer to Figure 7.117. Figure 7.117 shows a mechanical seal utilized for pump applications, while Figure 7.118 shows a mechanical seal utilized for compressor application. The seal designs appear to be almost identical. Close attention to Figure 7.118, however, will show reliefs of the rotating face of the seal. Considering that both seals operate on a fluid may give some hint as to why the designs are very similar. The objective of seal designs is to positively minimize leakage while removing frictional heat to obtain reliable continuous operation

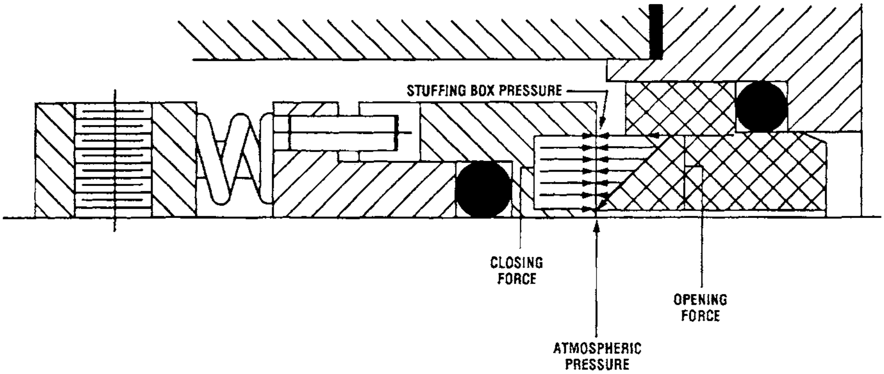


Figure 7.117 Typical pump single mechanical seal

of the seal. In a liquid application, the heat is removed by the fluid which passes between the rotating and stationary faces and changes from a liquid to gaseous state (heat of vaporization). This is precisely why all seals are said to leak and explains the recent movement in the industry to sealless pumps in toxic or flammable service. If the fluid between the rotating faces now becomes a gas, its capacity to absorb frictional heat is significantly less than that of a liquid. Therefore an 'equivalent orifice' must continuously exist between the faces to reduce friction and allow a sufficient amount of fluid to pass and thus take away the heat. The problem obviously is how to obtain this 'equivalent orifice'. There are many different designs of gas seals. However, regardless of the design, the dynamic action of the rotating face must create a dynamic force that will overcome the static forces acting on the seal to create an opening and hence 'equivalent orifice'.

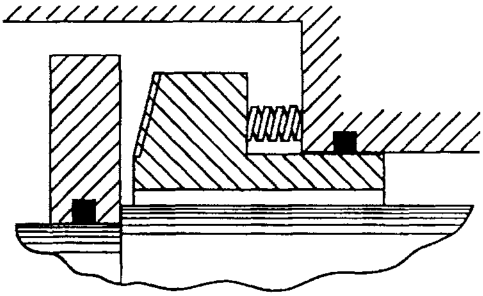


Figure 7.118 Typical design for curved face – spiral groove non-contact seal; curvature may alternately be on rotor

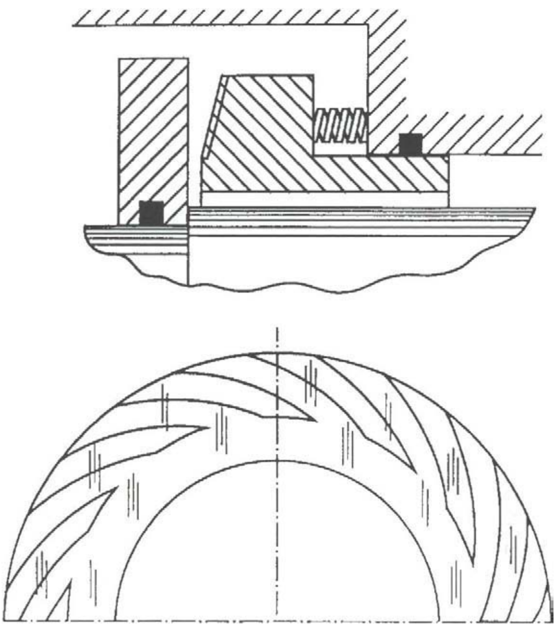


Figure 7.119 Dry gas seal. Top: typical design for curved face – spiral groove non-contact seal; curvature may alternately be on rotor; Bottom: Typical spiral groove pattern on face of seal typical non-contact gas seal (Courtesy of John Crane Co.)

Refer to Figure 7.119 which shows a typical dry seal. Notice the spiral grooves in this picture, they are typically machined at a depth of 100–400 micro inches. When rotating, these vanes create a high head low flow impeller that pumps gas into the area between the stationary and the rotating face, thereby increasing the pressure between the faces. When this pressure is greater than the static pressure holding the faces together, the faces will separate thus forming an equivalent orifice. In this specific seal design, the annulus below the vanes forms a tight face such that under static (stationary) conditions, zero leakage can be obtained if the seal is properly pressure balanced. Refer to Figure 7.120 for a force diagram that shows how this operation occurs.

Ranges of operation

Essentially, gas seals can be designed to operate at speeds and pressure differentials equal to or greater than those of liquid seals. Present state-of-the-art limits seal face differentials to approximately 1,000–1,500 psi and rubbing speeds to approximately 400 feet per second. Temperatures of operation can reach as high as 1,000°F. Where seal face differential exceeds these values, seals can be used in series (tandem) to meet specifications provided sufficient axial space is available in the seal housing.

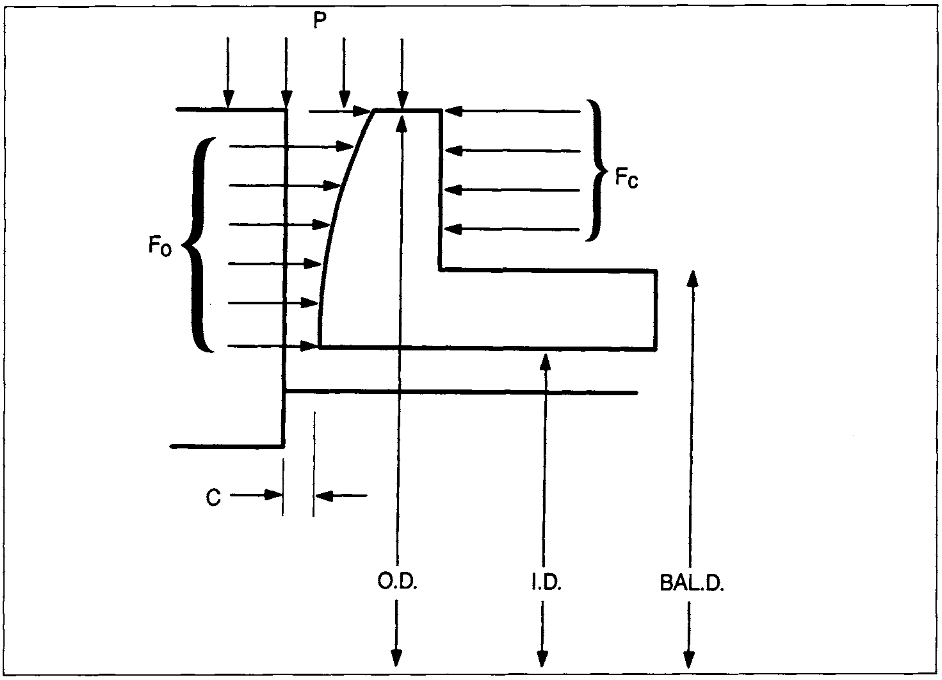


Figure 7.120 Hydrostatic force balance on seal stator ($F_C = F_O$) (Courtesy of John Crane Co.)

Leakage rates

Since the gas seal when operating forms an equivalent orifice, whose differential is equal to the supply pressure minus the seal reference pressure, there will always be a certain amount of leakage. Refer to Figure 7.121 for leakage graphs.

It can be stated in general that for most compressor applications with suction pressures on the order of 500 psi and below, leakage can be maintained on the order of one standard cubic foot per minute per seal. For a high pressure application (1,000–2,000 psi), differential leakage values can be as high as five standard cubic feet per minute per seal. As in any seal design, the total leakage is equal to the leakage across the seal faces and any leakage across secondary seals (O-rings, etc.). There have been reported incidence of explosive O-ring failure on rapid decompression of systems incorporating gas seals, thus resulting in excessive leakage. Consideration must be given to the system in order to tailor system decompression times in order to meet the requirements of the secondary seals. As previously mentioned, all gas seals will leak, but not until the face 'lifts off'. This speed known, oddly enough, as 'lift off speed' is usually less than 500 rpm. Caution must be exercised in variable speed applications to assure the system prevents the operation of the variable speed driver below this minimum lift off value.

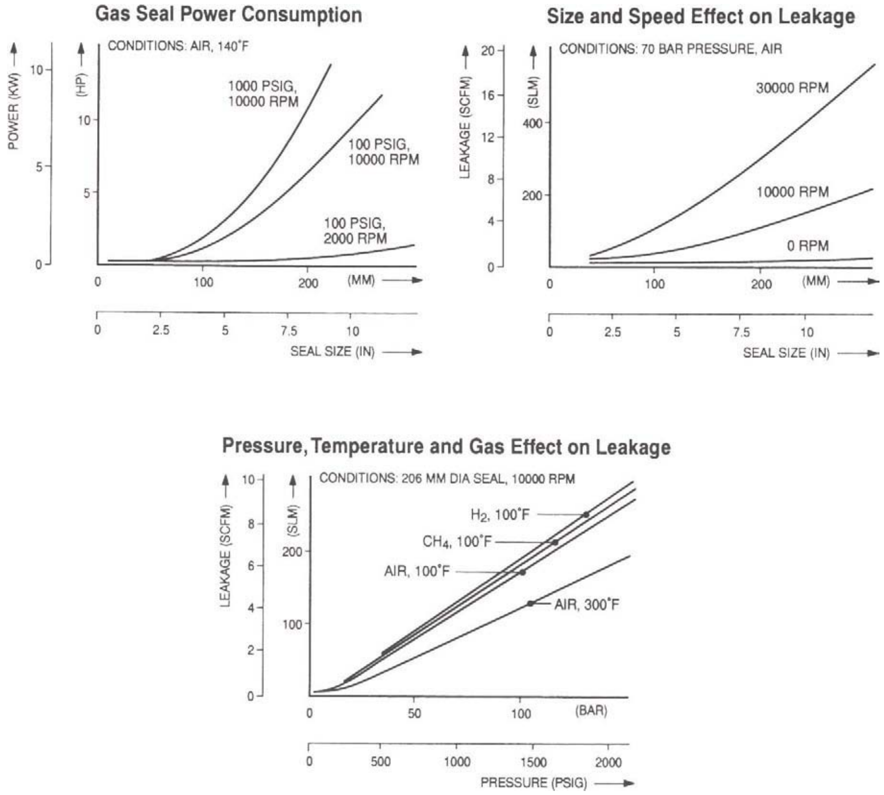


Figure 7.121 Dry gas seal leakage rates (Courtesy of John Crane Co.)

One recommendation concerning instrumentation is to provide one or two thermocouples in the stationary face of each seal to measure seal face temperature. This information is very valuable in determining lift off speed and condition of the grooves in the rotating seal face. Any clogging of these grooves will result in a higher face temperature and will be a good indication of requirement for seal maintenance.

Gas seal system types

As mentioned in this section, in order to assure the safety and reliability of gas seals, the system must be properly specified and designed. Listed below are but a few typical gas seal systems.

Low/medium pressure applications – air or inert gas

Figure 7.115 shows such a system. This system is identical to that of a liquid pump flush system incorporating relatively clean fluid that meets the requirements of the seal in terms of temperature and pressure. This system takes the motive fluid from the discharge of the compressor through dual filters (ten microns or less) incorporating a differential pressure gage and proportions equal flow through flow meters to each seal on the compressor. As previously discussed, compressors are usually pressure balanced such that the pressure on each end is approximately equal to the suction pressure of the compressor. The clean gas then enters the seal chamber and has two main paths:

A Through the internal labyrinth back to the compressor.

B Across the seal face and back to either the suction of the compressor or to vent.

Since the gas in this application is inert, it can be vented directly to the atmosphere or can be put back to the compressor suction. This would be the case also for a flammable gas. It must be noted, however, that this port is next to the journal bearing. Therefore a means of positively preventing entry of lube oil into this port must be provided in order to prevent the loss of lube oil or prevent the ingestion of lube oil into the compressor if this line is referenced back to the compressor suction. A suitable design must be incorporated for this bushing. Typically called a disaster bushing, it serves a dual purpose of isolating the lube system from the seal system and providing a means to minimize leakage of process fluid into the lube system in the event of a gas seal failure. In this system, a pressure switch upstream of an orifice in a flare line is used as an alarm and a shutdown to monitor flow. This switch uses the concept of an equivalent vessel in that increased seal leakage will increase the rate of supply versus demand flow in the equivalent vessel (pipe) and result in a higher pressure. When a high flow is reached, the orifice and pressure switch setting are thus sized and selected to alarm and shut down the unit if necessary. As in any system, close attention to changes in operating parameters is required. Flow meters must be properly sized and maintained clean such that relative changes in the flows can be detected in order to adequately plan for seal maintenance.

High pressure applications – air or inert gas

In this application, for pressures in excess of 1,000 psi, a tandem seal arrangement or series seal arrangement is usually used. Since failure of the inner seal would cause significant upset of the seal system, and large amounts of gas escaping to the atmosphere, a backup seal is employed. Refer to Figure 7.122.

The arrangement is essentially the same as low/medium pressure applications except that a backup seal is used in place of the disaster bushing. Most designs still incorporate a disaster bushing between the backup seal and the bearing cavity. Attention in this design must be

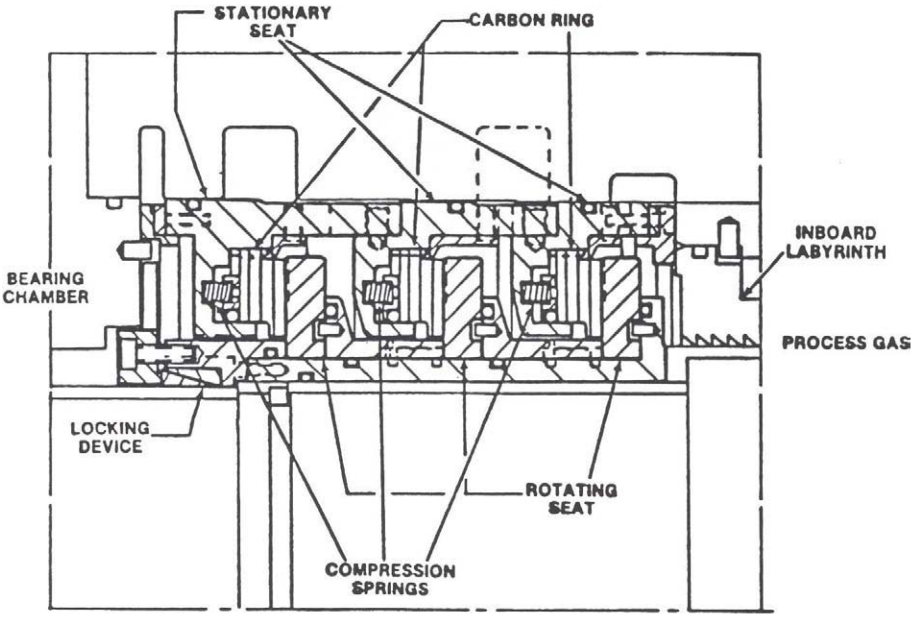


Figure 7.122 Dry gas seal: tandem dry gas seal arrangement (Courtesy of Dresser-Rand Corp.)

given to control of the inter-stage pressure between the primary and backup seal. Experience has shown that low differentials across the backup seal can significantly decrease its life. As in the case of liquid seals, a minimum pressure in the cavity between the seals of 25–30 psi is usually specified. This is achieved by properly sizing the orifice in the vent or reference line back to the suction to assure this pressure is maintained. All instrumentation and filtration are identical to that of the previous system.

Low/medium pressure application toxic or flammable gas

Please refer to Figure 7.123.

This system incorporates a double mechanical seal. In order to eliminate the possibility of flammable or toxic process gas escaping to the atmosphere, a buffer gas system utilizing inert gas is constantly injected between the primary and secondary seal. The buffer gas is regulated such that the pressure at the primary seal interface is greater than the seal process gas pressure. In order to achieve this objective, a differential control valve is installed on the inlet to the buffer gas port which senses pressure inside the cavity between the seals and reference pressure on the compressor side of the primary seal. A pressure differential of three to five psi is maintained to assure flow of buffer gas

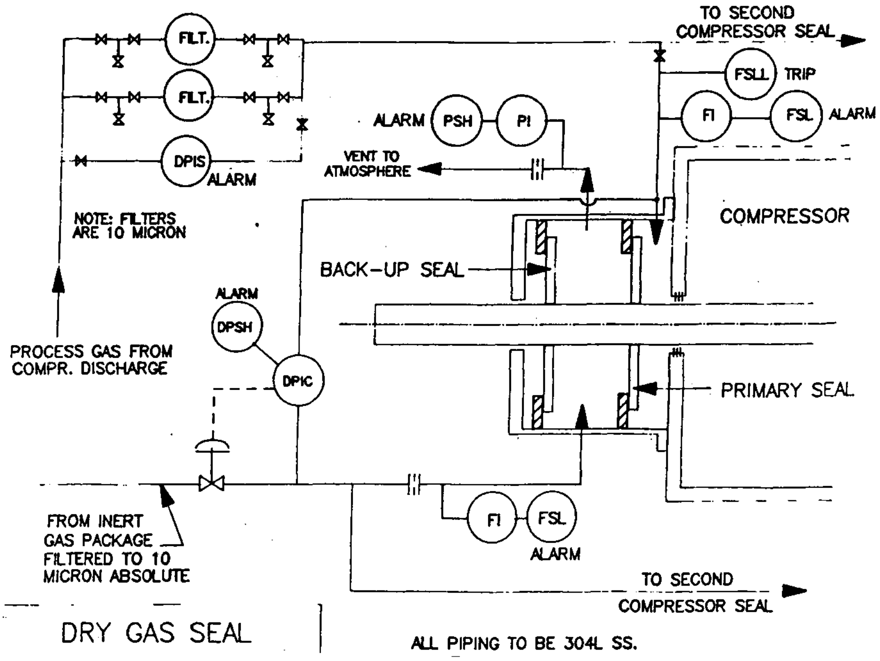

TO SECOND COMPRESSOR SEAL
FLT.
FLT.
DPIS
ALARM
NOTE: FILTERS ARE 10 MICRON
ALARM
PSH
PI
VENT TO ATMOSPHERE
FSLL
TRIP
FI
FSL
ALARM
COMPRESSOR
BACK-UP SEAL
ALARM
DPSH
DPIC
PROCESS GAS FROM COMPR. DISCHARGE
PRIMARY SEAL
FROM INERT GAS PACKAGE FILTERED TO 10 MICRON ABSOLUTE
FI
FSL
ALARM
TO SECOND COMPRESSOR SEAL
DRY GAS SEAL
ALL PIPING TO BE 304L SS.

Figure 7.123 Double gas seal configuration for flammable or toxic process gas service (Courtesy of M.E. Crane Consultant)

to the unit. The backup seal then seals between the pressure between the two seals, which is now inert gas, and the atmosphere thus creating a safe seal situation at all times.

In this design, however, absence of proper buffer gas flow (differential between seal faces) will be cause for alarm on decreasing differential and unit shutdown if the condition does not correct itself. The reliability of this system is dependent on the buffer gas system. This means that the buffer gas system must continuously supply the buffer fluid at the correct pressure, temperature, flow rate, etc. A buffer gas compressor shutdown or failure will necessitate shutdown of the critical equipment. The buffer gas compressor could range from a large nitrogen compressor down to a small packaged unit furnished with a buffer gas inert package. Inert gas packages that generate nitrogen directly from atmospheric air are available to utilize as a buffer gas source for this design where a large nitrogen system is not available.

A variation of this design would have to be considered if the process did not allow leakage of inert gas into the process system. This would be the case in any process recycle application such as a hydrogen reformer, etc. In this case, a double seal would be utilized as the primary seal and would seal between suction and discharge pressure. The discharge flow would be filtered as in the case of example one in this section. Leaks from the primary seal into the port between the seals would then be washed with inert gas. The backup seal would be designed such that buffer gas would be admitted to the inner cavity via a tight labyrinth and leakage would be controlled by an orifice to vent or flare. The design would be such that the ratio of inner to process gas flow would be approximately ten to one. The backup seal leakage of inert gas to atmosphere would be controlled tightly by the seal gas face. In this design, flow meters would be typically used to measure maximum flow from the inert gas system. Excess flow would initiate an alarm and final shutdown of the unit.

There are different schools of thought concerning the use of buffer gasses. In pipeline operations, frequently the gas between the primary and backup seal may be sweet fuel gas. In this case, caution must be paid to failure of the backup seal. Failure of the backup seal will admit sweet fuel gas directly to the atmosphere and the lube oil system.

Summary

Since there are significant advantages to the use of dry gas seals, many units are being retrofitted in the field which incorporates this system. In many cases, significant payouts can be realized.

If a unit is to be retrofitted, it is strongly recommended that the design of the gas seal be thoroughly audited to assure safety and reliability. As mentioned in this section, retrofitting from a liquid to a gas seal system renders the unit a separate system type unit, that is, a separate lube and gas seal system. Naturally, loss of lube oil into the seal system will result in significant costs and could result in seal damage or failure by accumulating debris between the seal rotating and the stationary faces. The adequate design of the separation barriers between the lube and seal face must be thoroughly examined and audited to assure reliable and safe operation of this system. Many unscheduled field shutdowns and safety situations have resulted from the improper design of the lube system, seal system separation labyrinth. In addition to the above considerations, a critical speed analysis, rotor response and stability analysis (if applicable) should always be conducted when retrofitting from liquid to dry gas seals.

Auxiliary system types and functions

(Auxiliary systems knowledge)

- Introduction
- Types of auxiliary systems
- Summary of system functions, similarities and differences

Introduction

In this chapter, we will direct our attention to the types of auxiliary systems that will be covered. The major types of systems to be covered will be overviewed and the functions, similarities and differences of each system type will be summarized. Note that only the functional design as detailed on the system schematic will be discussed. The system arrangement design as detailed on the system outline or model will be discussed in a later chapter.

Types of auxiliary systems

The following is a brief summary of the types of auxiliary systems that will be covered. It must be understood that there are many variations of these systems both in terms of functional design and arrangement of components. However, the basic function of a specific type of system is the same regardless of the variations. As we proceed through the course, many of the system variations will be presented. Readers are encouraged to introduce system variations with which they are familiar. All of the system types presented in this section will be covered in detail later in this book.

Lubrication systems

There are about as many different types of lubrication systems as there are lubricants. In this course we will confine our attention to the pressurized types that should always be used with critical equipment. Two types of pressurized lube oil systems using positive displacement pumps are shown in Figures 7.124 and 7.125. Figure 7.124 depicts a system with a shaft driven lube oil pump while Figure 7.125 shows both pumps driven by sources other than the shaft. Figure 7.126 details a lubrication system using dynamic (centrifugal) pumps.

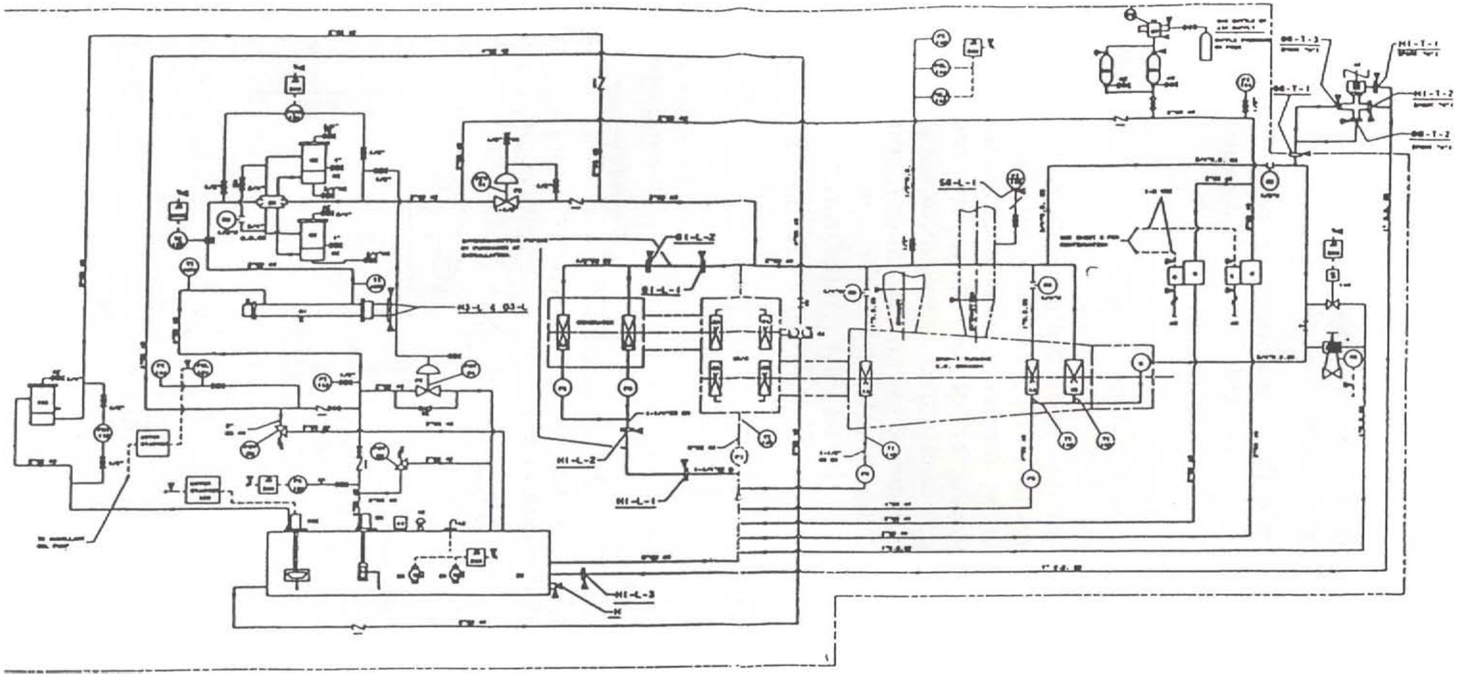

HI-T-1
HI-T-2
SE-L-1
SI-L-2
SI-L-1
HI-L-2
HI-L-1
HI-L-3

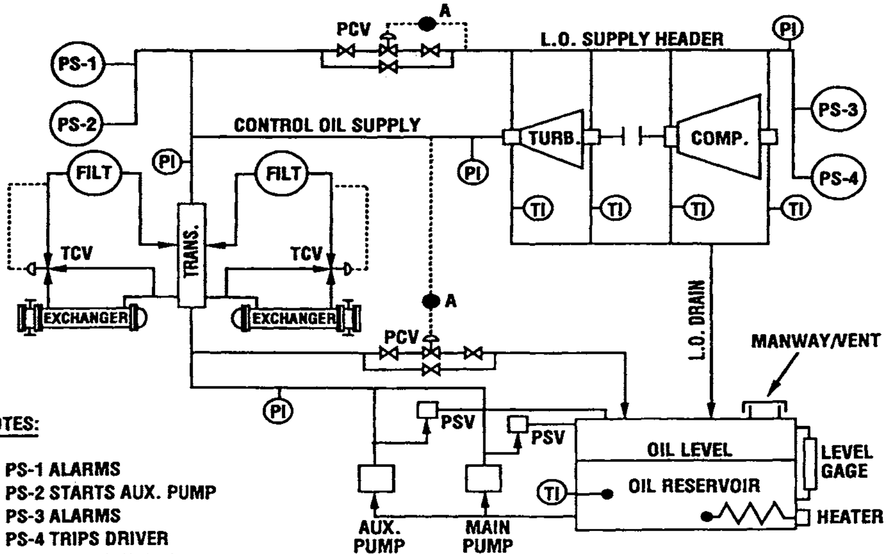
Figure 7.124 Lube oil system main pump shaft driven (Courtesy of Elliott Co.)

Regardless of the design variations or component type, the basic function of any lubrication system in critical equipment service is as follows:

To *continuously supply clean lubricating fluid* to *each specified point* at the required *pressure, temperature* and *flow rate.*

As an exercise, identify the component or components in Figures 7.124, 7.125 and 7.126 that are required to satisfy each italicized phrase in the above definition.

Before we proceed, one comment concerning the types of pumps used in auxiliary systems. The performance of positive displacement pumps (screw, gear, rotary, piston, diaphragm, etc.) is not significantly affected by the characteristics of the liquid pumped. However, dynamic pump (centrifugal) performance varies significantly with different liquid characteristics. As a result, dynamic pumps are used where the pumped liquid characteristics can easily be controlled. Since the liquid characteristics (viscosity) of oil varies considerably with temperature and the oil reservoir temperature is not accurately controlled, positive displacement pumps are usually used for systems containing oil. However, in cases where the oil inlet temperature to the pump can be held relatively constant (±25°F), a dynamic type pump can be used.


PCV
A
L.O. SUPPLY HEADER
PI
PS-1
PS-2
CONTROL OIL SUPPLY
TURB.
COMP.
PS-3
PS-4
FILT
PI
FILT
PI
TI
TI
TI
TI
TCV
TRANS.
TCV
EXCHANGER
EXCHANGER
A
L.O. DRAIN
MANWAY/VENT
PCV
PI
PSV
PSV
OIL LEVEL
OIL RESERVOIR
TI
LEVEL GAGE
HEATER
AUX. PUMP
MAIN PUMP

NOTES:

1. **PS-1 ALARMS**
2. **PS-2 STARTS AUX. PUMP**
3. **PS-3 ALARMS**
4. **PS-4 TRIPS DRIVER**
5. **A = SNUBBER VALVE**

Figure 7.125 Typical lube oil supply system

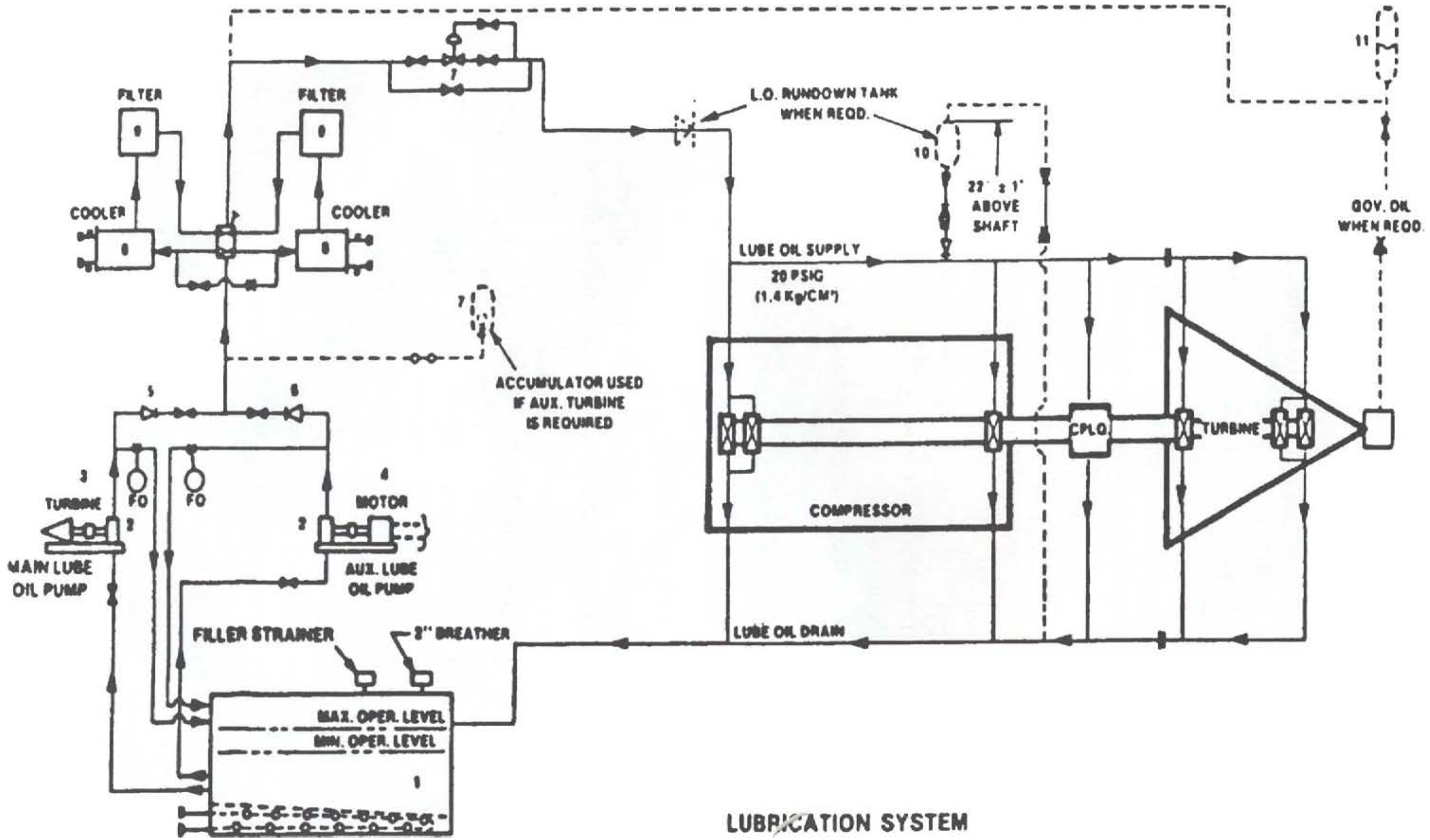

FILTER
FILTER
COOLER
COOLER
7
L.O. RUNDOWN TANK WHEN REQD.
10
22" ± 1" ABOVE SHAFT
11
GOV. OIL WHEN REQD.
LUBE OIL SUPPLY
20 PSIG
(1.4 Kg/CM²)
7
ACCUMULATOR USED IF AUX. TURBINE IS REQUIRED
5
6
3
TURBINE
FO
FO
4
MOTOR
2
2
MAIN LUBE OIL PUMP
AUX. LUBE OIL PUMP
COMPRESSOR
CPLG
TURBINE
LUBE OIL DRAIN
FILLER STRAINER
2" BREATHER
MAX. OPER. LEVEL
MIN. OPER. LEVEL
1
LUBRICATION SYSTEM

Figure 7.126 Lubrication system (Courtesy of Dresser-Rand)

Fluid sealing systems

Again, there are numerous types of fluid seal systems. In fact, there are more variations than lubrication systems since the types of seals utilized, sealing fluids and sealing pressures vary widely. Notice that this section is entitled *fluid* seal systems as opposed to liquid seal systems. A fluid by definition can be a liquid or a vapor. There are both liquid and vapor (gas) seal systems. Regardless of the type, the function of a critical equipment seal system is:

> To *continuously supply clean fluid* to *each specified seal interface point* at the required *differential pressure, temperature* and *flow rate.*

Naturally, the function of the seal is to contain the process fluid. Since the seal interface is between the rotating and stationary components of the seal, friction and corresponding energy loss (BTU's/#, horsepower) result. The seal fluid (oil, water, vapor) must be introduced into the seal at the correct differential pressure above the process fluid pressure in order to contain the process fluid and to remove the heat of seal interface friction.

Three types of seal systems are presented in this section for turbo-compressor applications:

- Figure 7.127 – Seal oil system – contact seal
- Figure 7.128 – Seal oil system – bushing seal
- Figure 7.129 – Gas seal system

As an exercise, compare the seal system components with the lubrication system components and state the major differences.

Figure 7.130 shows a combined lubrication and sealing system that would be used with a liquid seal. Since the function of a lubrication and a seal system are similar they can be combined in certain instances where required flow rates, pressures and *process gas characteristics* permit. Extreme care must be exercised when designing a combined system containing entrained seal gas in the sealing oil (process gas or inert-buffer gas). Since the system is combined, the seal oil leakage, if it is returned to the reservoir, must be effectively 'degassed' to assure the bearing lubricating film is incompressible (gas free). Failure to do this can result in catastrophic bearing and equipment failure. In addition, a saturated lube oil will have a significantly lower 'flashpoint'. The flashpoint is defined as the temperature at which a lube oil will support combustion. There have been cases where returned seal oil leakage flashpoints have been recorded as low as 120°F! (Typical oil flashpoints are 400–500°F.) Seal oil leakage degassing is a primary factor in seal oil system safety and reliability.

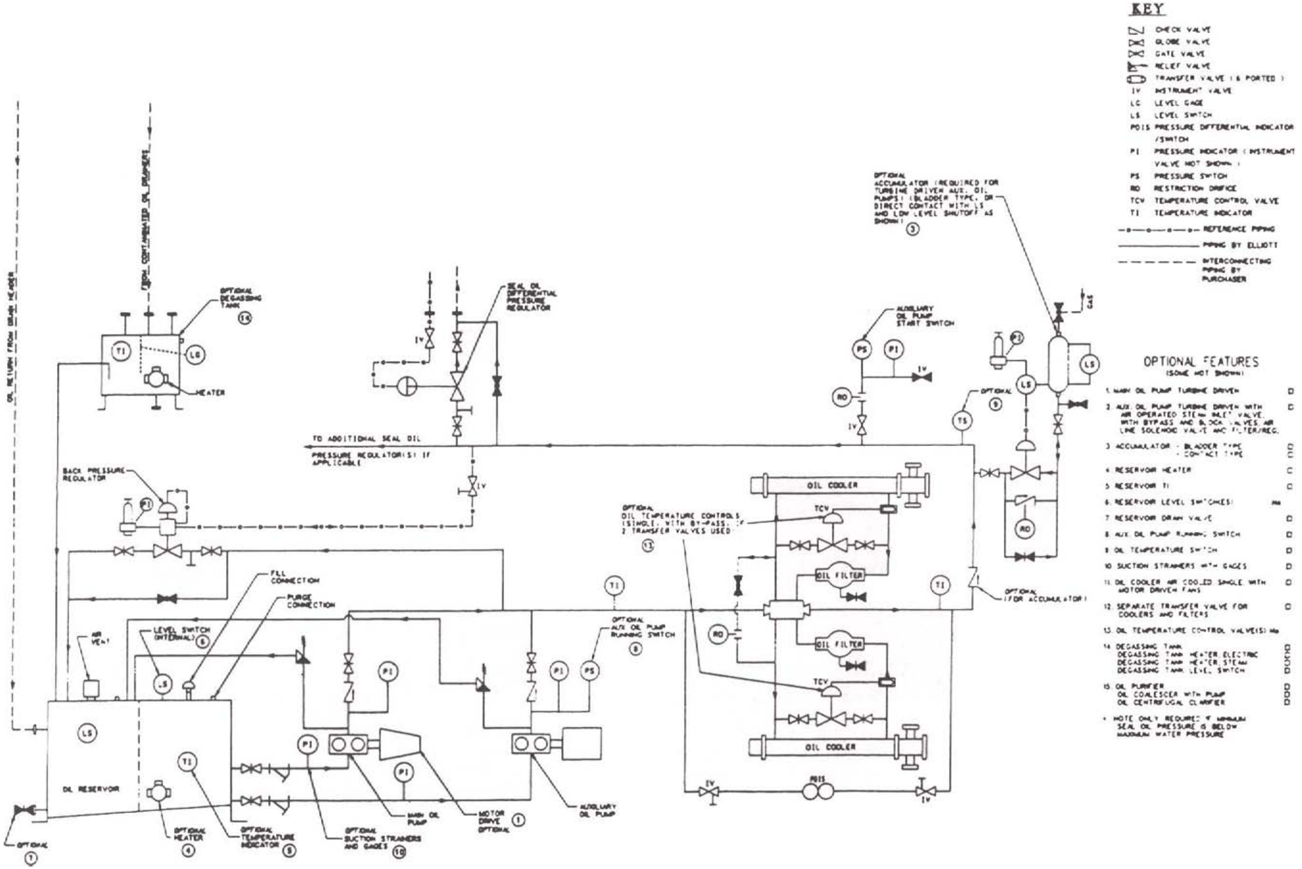

KEY
CHECK VALVE
GLOBE VALVE
GATE VALVE
RELIEF VALVE
TRANSFER VALVE (6 PORTED)
IV INSTRUMENT VALVE
LG LEVEL GAGE
LS LEVEL SWITCH
PDIS PRESSURE DIFFERENTIAL INDICATOR /SWITCH
PI PRESSURE INDICATOR (INSTRUMENT VALVE NOT SHOWN)
PS PRESSURE SWITCH
RO RESTRICTION ORIFICE
TCV TEMPERATURE CONTROL VALVE
TI TEMPERATURE INDICATOR
REFERENCE PIPING
PIPING BY ELLIOTT
INTERCONNECTING PIPING BY PURCHASER
OPTIONAL FEATURES
(SOME NOT SHOWN)
1. MAIN OIL PUMP TURBINE DRIVEN ☐
2. AUX. OIL PUMP TURBINE DRIVEN WITH AIR OPERATED STEAM INLET VALVE, WITH BYPASS AND BLOCK VALVES, AIR LINE SOLENOID VALVE AND FILTER/REG. ☐
3. ACCUMULATOR - BLADDER TYPE ☐
- CONTACT TYPE ☐
4. RESERVOIR HEATER ☐
5. RESERVOIR TI ☐
6. RESERVOIR LEVEL SWITCH(ES) NA
7. RESERVOIR DRAIN VALVE ☐
8. AUX. OIL PUMP RUNNING SWITCH ☐
9. OIL TEMPERATURE SWITCH ☐
10. SUCTION STRAINERS WITH GAGES ☐
11. OIL COOLER AIR COOLED SINGLE WITH MOTOR DRIVEN FANS ☐
12. SEPARATE TRANSFER VALVE FOR COOLERS AND FILTERS ☐
13. OIL TEMPERATURE CONTROL VALVE(S) NA
14. DEGASSING TANK ☐
DEGASSING TANK HEATER, ELECTRIC ☐
DEGASSING TANK HEATER, STEAM ☐
DEGASSING TANK LEVEL SWITCH ☐
15. OIL PURIFIER ☐
OIL COALESCER WITH PUMP ☐
OIL CENTRIFUGAL CLARIFIER ☐
* NOTE ONLY REQUIRED IF MINIMUM SEAL OIL PRESSURE IS BELOW MAXIMUM WATER PRESSURE
OIL RETURN FROM DRAIN HEADER
FROM CONTAMINATED OIL DRAINERS
OPTIONAL DEGASSING TANK ⑭
TI
LG
HEATER
SEAL OIL DIFFERENTIAL PRESSURE REGULATOR
IV
TO ADDITIONAL SEAL OIL PRESSURE REGULATOR(S) IF APPLICABLE
BACK PRESSURE REGULATOR
PI
IV
FILL CONNECTION
PURGE CONNECTION
LEVEL SWITCH (INTERNAL) ⑥
AIR VENT
LS
LS
TI
OIL RESERVOIR
PI
PI
PI
PI
PS
OPTIONAL AUX OIL PUMP RUNNING SWITCH ⑧
MAIN OIL PUMP
MOTOR DRIVE OPTIONAL ①
AUXILIARY OIL PUMP
OPTIONAL ⑦
OPTIONAL HEATER ④
OPTIONAL TEMPERATURE INDICATOR ⑤
OPTIONAL SUCTION STRAINERS AND GAGES ⑩
OPTIONAL OIL TEMPERATURE CONTROLS (SINGLE, WITH BY-PASS, IF 2 TRANSFER VALVES USED) ⑬
TI
RO
OIL COOLER
TCV
OIL FILTER
OIL FILTER
TCV
OIL COOLER
TI
IV
PDIS
IV
OPTIONAL ACCUMULATOR (REQUIRED FOR TURBINE DRIVEN AUX. OIL PUMPS) (BLADDER TYPE, OR DIRECT CONTACT WITH LS AND LOW LEVEL SHUTOFF AS SHOWN) ③
AUXILIARY OIL PUMP START SWITCH
PS
PI
IV
RO
IV
TS
OPTIONAL ⑨
PI
LS
LS
GAS
RO
OPTIONAL (FOR ACCUMULATOR)

Figure 7.127 Elliott seal oil console schematic for compressors with ISO-carbon seals (Courtesy of Elliott Co)

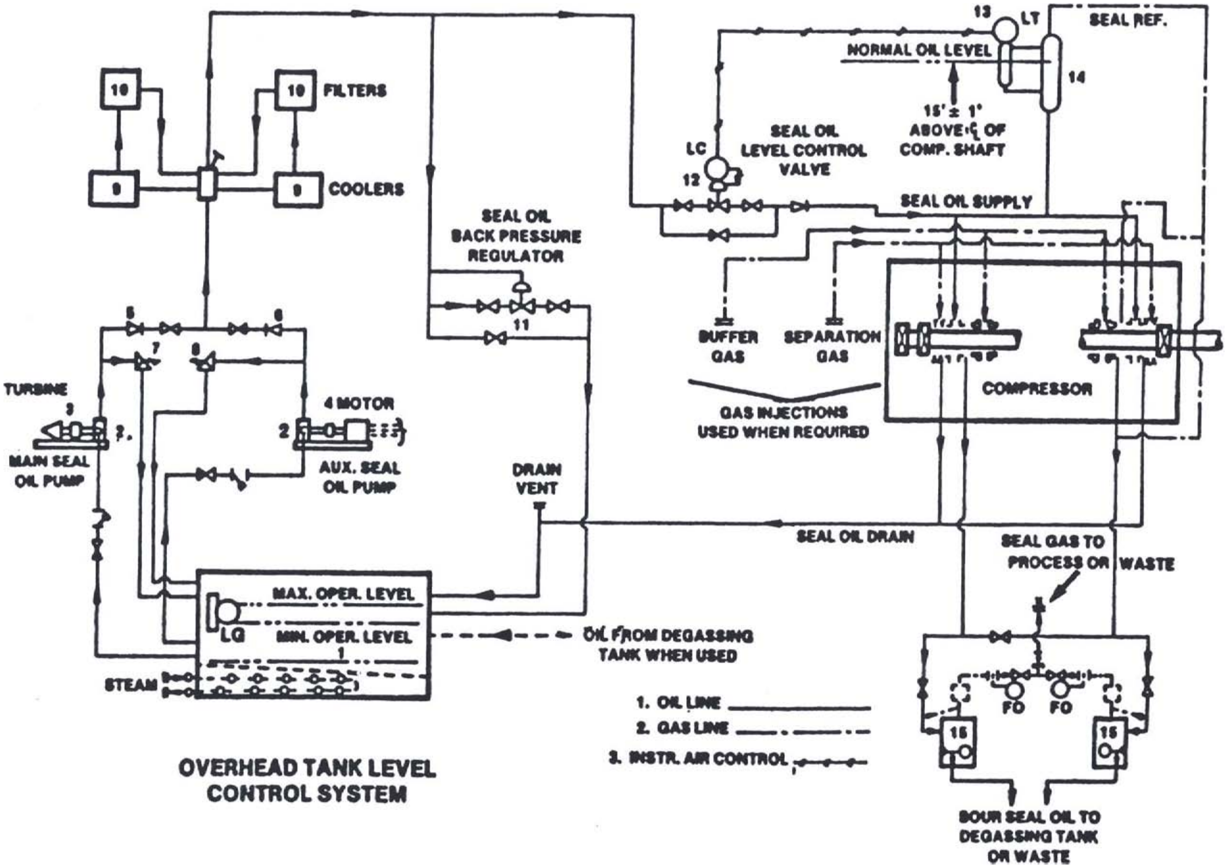

13
LT
SEAL REF.
NORMAL OIL LEVEL
14
15' ± 1'
ABOVE ℄ OF
COMP. SHAFT
10
10
FILTERS
9
9
COOLERS
LC
12
SEAL OIL
LEVEL CONTROL
VALVE
SEAL OIL SUPPLY
SEAL OIL
BACK PRESSURE
REGULATOR
11
5
6
7
8
TURBINE
3
4 MOTOR
2
2
MAIN SEAL
OIL PUMP
AUX. SEAL
OIL PUMP
BUFFER
GAS
SEPARATION
GAS
COMPRESSOR
GAS INJECTIONS
USED WHEN REQUIRED
DRAIN
VENT
SEAL OIL DRAIN
SEAL GAS TO
PROCESS OR WASTE
MAX. OPER. LEVEL
LG
MIN. OPER. LEVEL
1
STEAM
OIL FROM DEGASSING
TANK WHEN USED
FO
FO
15
15
1. OIL LINE
2. GAS LINE
3. INSTR. AIR CONTROL
OVERHEAD TANK LEVEL
CONTROL SYSTEM
SOUR SEAL OIL TO
DEGASSING TANK
OR WASTE

Figure 7.128 Bushing seals (Courtesy of Dresser-Rand)

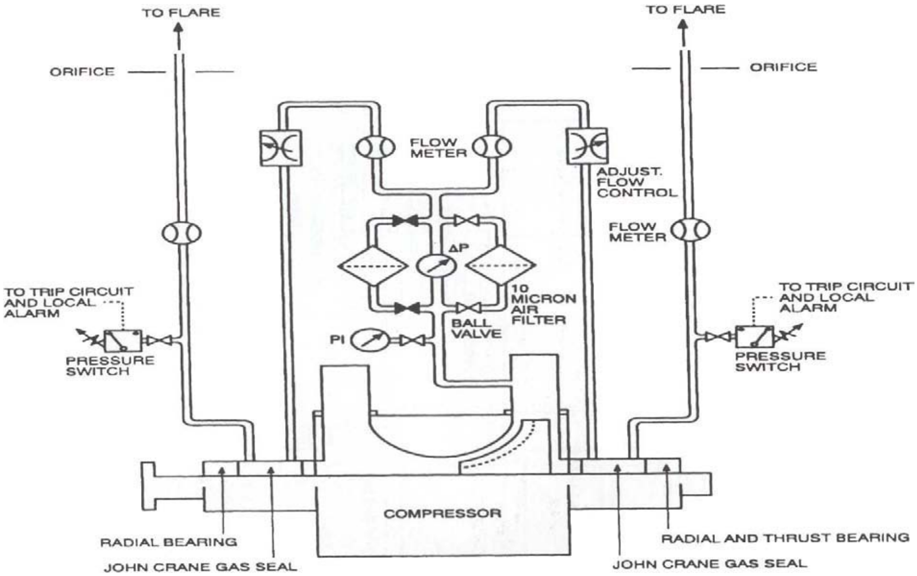

TO FLARE
TO FLARE
ORIFICE
ORIFICE
FLOW METER
ADJUST. FLOW CONTROL
FLOW METER
ΔP
TO TRIP CIRCUIT AND LOCAL ALARM
10 MICRON AIR FILTER
TO TRIP CIRCUIT AND LOCAL ALARM
BALL VALVE
PI
PRESSURE SWITCH
PRESSURE SWITCH
COMPRESSOR
RADIAL BEARING
JOHN CRANE GAS SEAL
RADIAL AND THRUST BEARING
JOHN CRANE GAS SEAL

Figure 7.129 Typical gas seal system (Courtesy of John Crane Co)

Buffer fluid systems

This large category of auxiliary systems covers buffer gas systems for turbo-compressors, steam seal systems, eductor systems and pump flush systems.

In turbo-compressor applications, a buffer fluid (gas) can be used for any of the following purposes:

A. Prevent the process gas from coming in contact with the sealing fluid

B. Prevent the sealing fluid from entering the process stream

C. Control the seal area environment (pressure and temperature)

Alternatives B and C above can use the process gas if it is available at the required pressures and temperatures and is compatible with the sealing fluid. Figure 7.131 is an example of a buffer gas system designed for alternative A.

Figure 7.132 shows a system designed for alternatives A and B.

Figure 7.133 shows a steam seal system whose purpose is to direct leak off steam to the specified locations thus preventing steam (moisture) leaking from the shaft seals and entering the lubrication system. This arrangement reduces the process stream to an acceptable pressure level and also uses a subatmospheric pressure source to educt air to prevent external seal steam leakage.

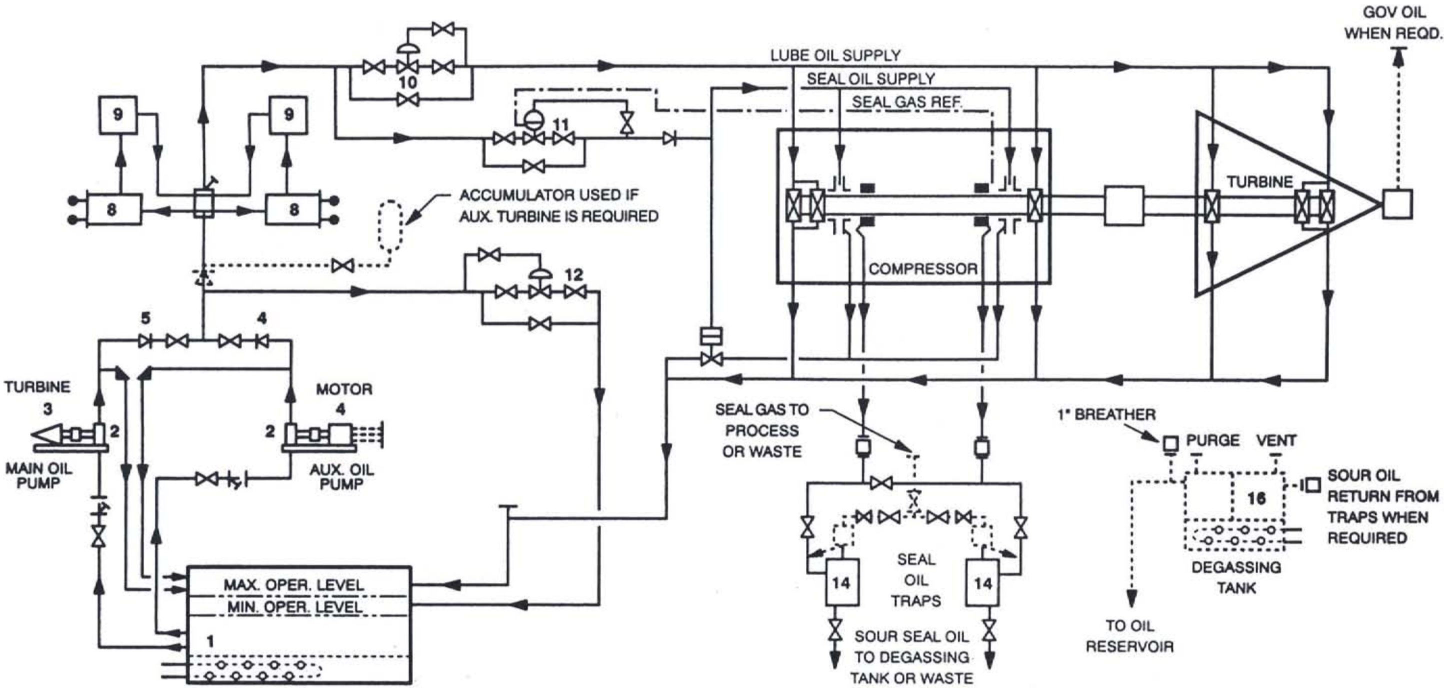

GOV OIL
WHEN REQD.
LUBE OIL SUPPLY
SEAL OIL SUPPLY
SEAL GAS REF.
9
9
10
11
8
8
ACCUMULATOR USED IF
AUX. TURBINE IS REQUIRED
TURBINE
COMPRESSOR
12
5
4
TURBINE
3
2
MAIN OIL
PUMP
MOTOR
4
2
AUX. OIL
PUMP
SEAL GAS TO
PROCESS
OR WASTE
1" BREATHER
PURGE
VENT
16
SOUR OIL
RETURN FROM
TRAPS WHEN
REQUIRED
DEGASSING
TANK
TO OIL
RESERVOIR
SEAL
OIL
TRAPS
14
14
SOUR SEAL OIL
TO DEGASSING
TANK OR WASTE
MAX. OPER. LEVEL
MIN. OPER. LEVEL
1

Figure 7.130 Combined lube and seal system (Courtesy of Dresser-Rand)

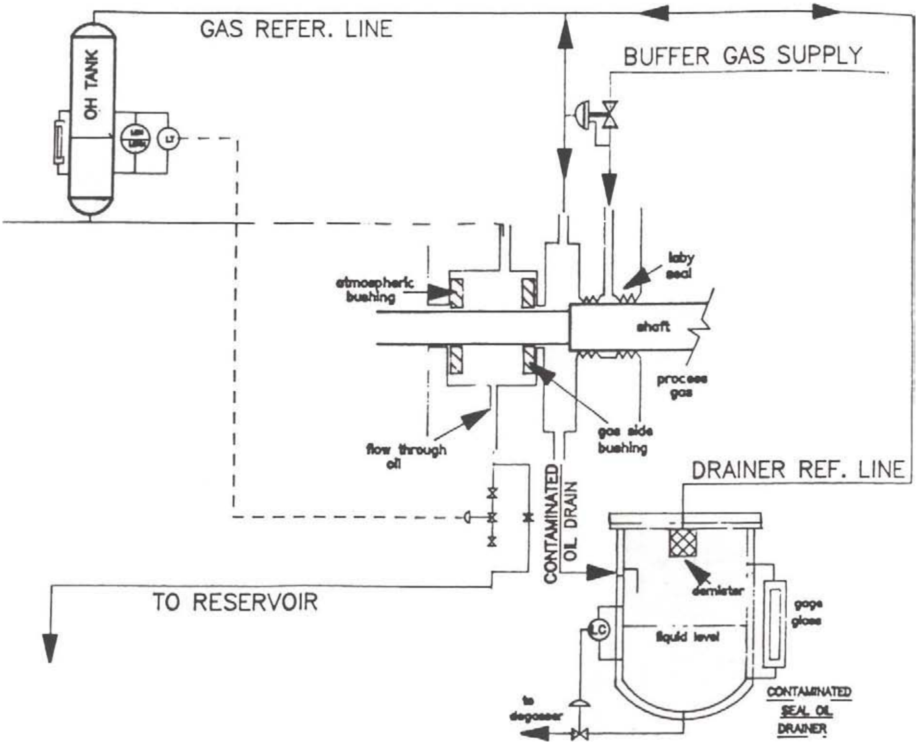

GAS REFER. LINE
BUFFER GAS SUPPLY
OH TANK
LT
laby seal
atmospheric bushing
shaft
process gas
flow through oil
gas side bushing
DRAINER REF. LINE
CONTAMINATED OIL DRAIN
TO RESERVOIR
demister
gage glass
LC
liquid level
to degasser
CONTAMINATED SEAL OIL DRAINER

Figure 7.131 Typical buffer gas system (Courtesy of M.E. Crane Consultant)

The final type of buffer fluid systems covered in this chapter involve pump sealing systems. When the pumped liquid conditions do not meet the requirements of the pump seal (pressure, temperature, cleanliness, liquid characteristics, etc.) a buffer system (commonly called a flush system) must be utilized. The buffer liquid can be the pumped liquid modified to meet the seal requirements (filtered or cooled), an externally supplied liquid compatible with the pumped liquid or a barrier liquid used in a tandem or double seal arrangement. Typical examples of these systems are shown in Figures 7.134, 7.135 and 7.136.

As in lubrication and sealing systems, the basic function of any critical equipment buffer fluid system is as follows:

> To *continuously supply clean buffer fluid* to *each specified point* at the required differential *pressure*, *temperature* and *flow rate*.

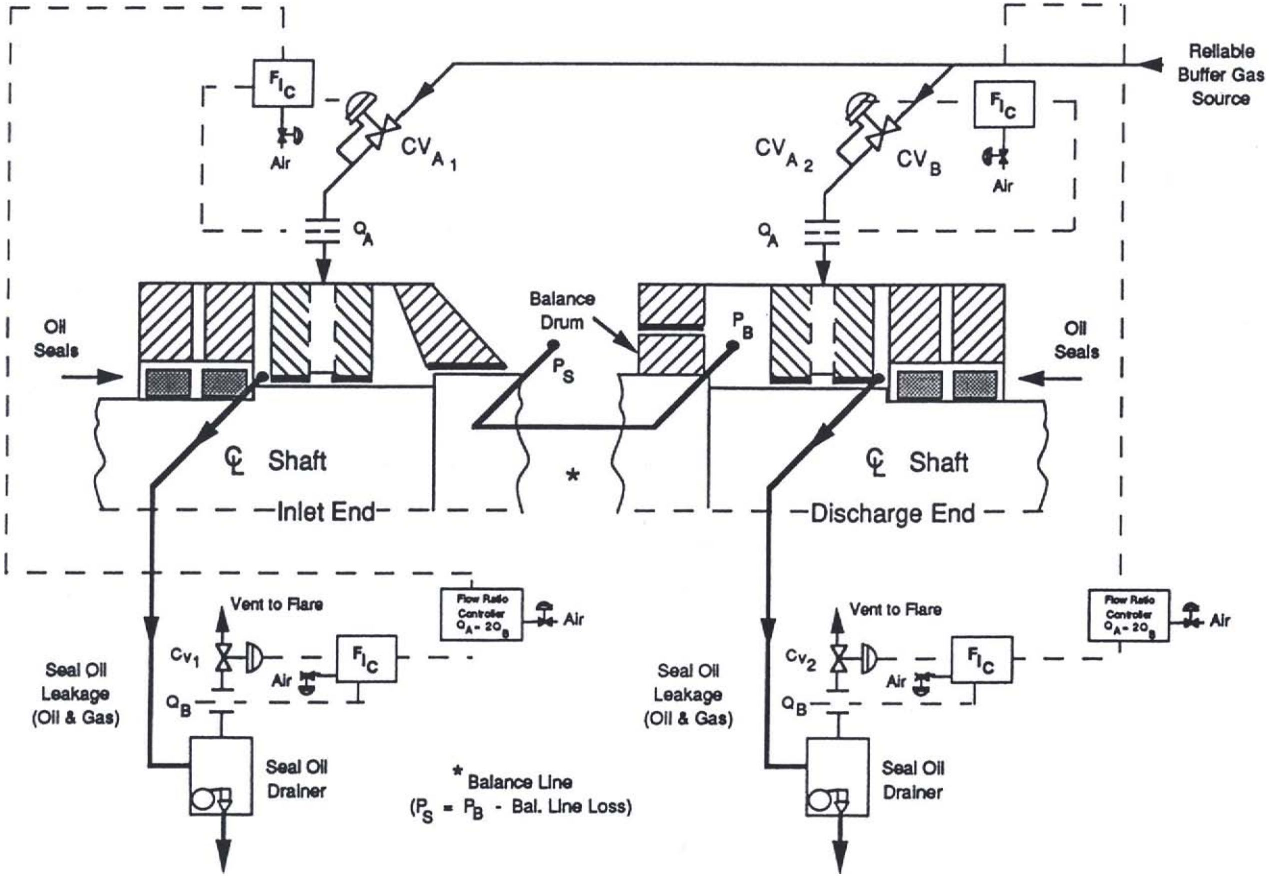

Reliable Buffer Gas Source
FIC
Air
CVA 1
CVA 2
CVB
QA
QA
Oil Seals
Oil Seals
Balance Drum
PB
PS
℄ Shaft
℄ Shaft
*
Inlet End
Discharge End
Vent to Flare
Vent to Flare
Flow Ratio Controller QA = 2QB
Flow Ratio Controller QA = 2QB
Cv1
Cv2
QB
QB
Seal Oil Leakage (Oil & Gas)
Seal Oil Leakage (Oil & Gas)
Seal Oil Drainer
Seal Oil Drainer
* Balance Line (PS = PB - Bal. Line Loss)

Figure 7.132 Flow ratio buffer gas control system

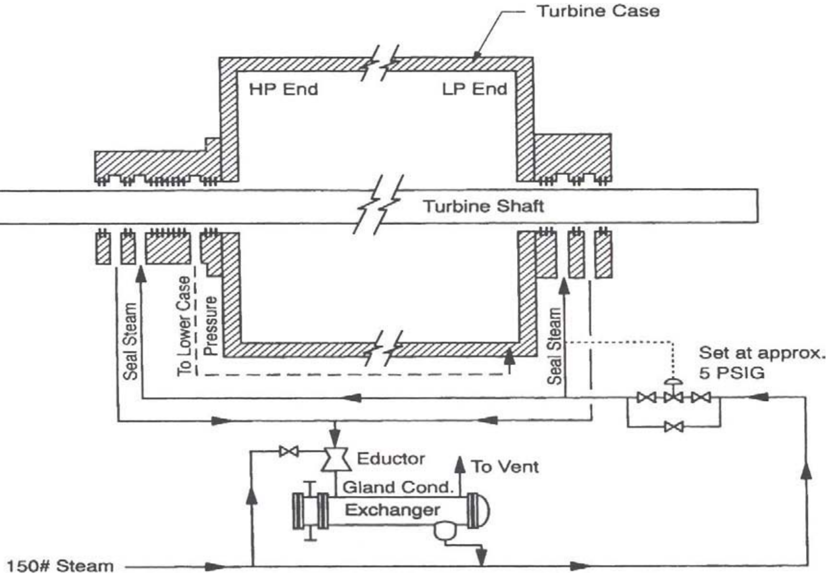

Turbine Case
HP End
LP End
Turbine Shaft
Seal Steam
To Lower Case Pressure
Seal Steam
Set at approx. 5 PSIG
Eductor
To Vent
Gland Cond. Exchanger
150# Steam

Figure 7.133 Steam turbine gland seal system (Courtesy of M.E. Crane Consultant)

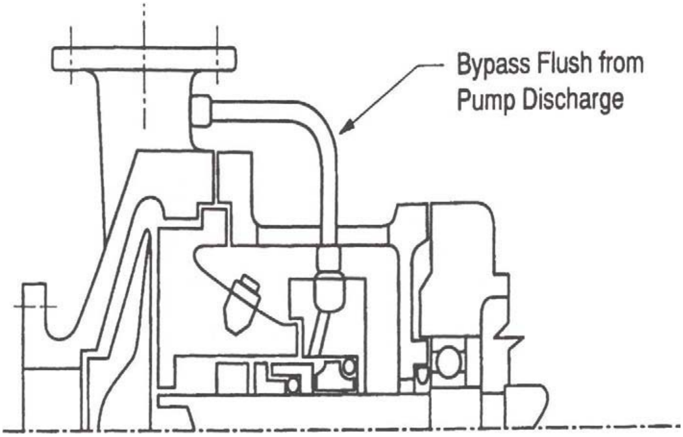

Bypass Flush from Pump Discharge

Figure 7.134 Single inside seal with bypass flush from pump discharge (Courtesy of Durametallic Corporation)

Plant utility systems

The purpose of including this topic is to bring attention to the fact that each item of critical equipment and all of its auxiliaries are part of the plant utility system which includes:

- The plant and instrument air system
- The plant steam system

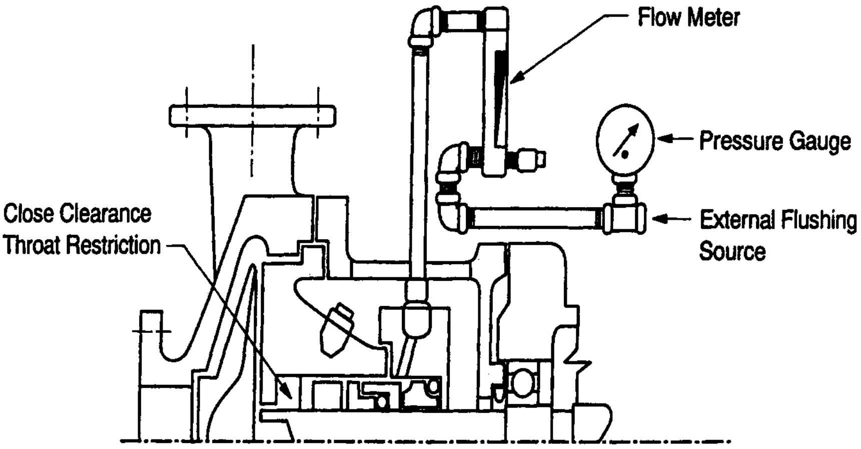

Flow Meter
Pressure Gauge
External Flushing Source
Close Clearance Throat Restriction

Figure 7.135 Single inside seal with flush from external source (Courtesy of Durametallic Corporation)

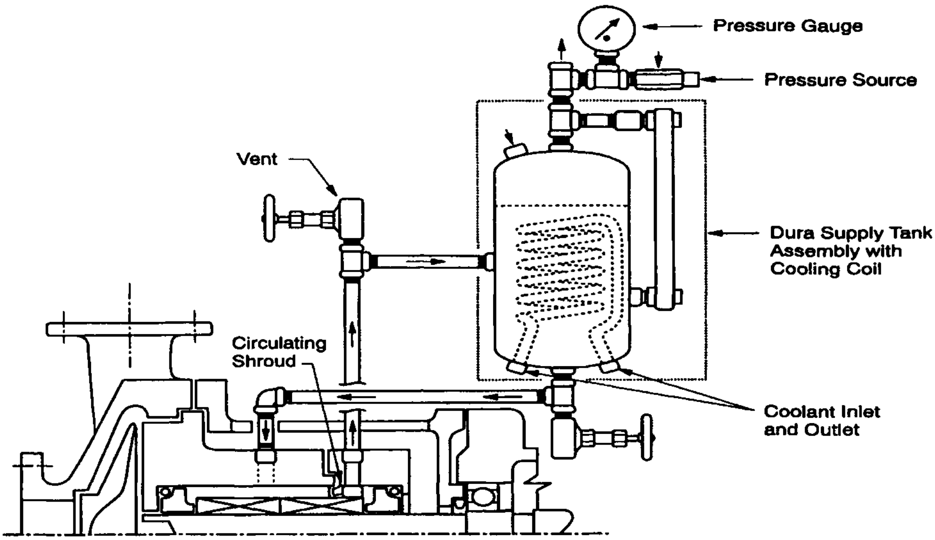

Pressure Gauge
Pressure Source
Vent
Dura Supply Tank Assembly with Cooling Coil
Circulating Shroud
Coolant Inlet and Outlet

Figure 7.136 Double inside seal with induced circulation through supply tank with cooling coil (Courtesy of Durametallic Corporation)

- The plant cooling water system

To name just a few. They are also auxiliary systems and have the same function:

> To *continuously supply clean utility fluid* to *each specified point* at the *required pressure, temperature and flow rate.*

Summary of system functions, similarities and differences

Having divided critical equipment auxiliary systems into four groups, and having defined the function of each group, let's attempt to create a universal auxiliary system function definition. Reviewing the stated function definition of each group and using only exactly similar phrases we obtain the following definition:

To continuously supply clean fluid to each specified point at the required pressure, temperature and flow rate.

The only major functional difference is that lubrication and plant utility systems require *pressure control* whereas seal and buffer systems require *differential pressure* control. Why is this so? Sealing and buffer systems require differential pressure control because the fluids they are acting on can vary in pressure but lubrication and utility systems ultimately act against atmospheric pressure which is essentially constant.

As one last section exercise let's examine the universal auxiliary system definition above and list the system components common to all auxiliary system groups.

It is hoped that the information contained in this section will enable the reader to better understand the function of each major group of auxiliary system types and how similar all the major auxiliary system groups are in terms of function. Once this fact is understood, the understanding of each specific type of auxiliary system becomes clearer.

Answers for each component and system function knowledge base section in this chapter

1. The rotor and the process system

Function exercise

Component: *Pump or compressor rotor*

I. 'What' - The function of Rotor is to *produce the specified value of fluid head and efficiency without impeller, shaft or any other installed component failure while maintaining acceptable values of machine vibration, axial displacement and bearing temperature.*

II. 'How' - The function of Rotor is accomplished by *proper impeller design tip speed and fluid velocity relative to the impeller and proper stationary passage design (volute, diffuser, stage seal and crossover). It is assumed that the all clearances are to specification, the rotor is balanced and the impeller and all passages are free of fouling. (Debris).*

Figure 7.137 Function exercise - Component: Pump or compressor rotor

Component failure causes

Component: *Pump or compressor rotor*

List all possible failure causes. (Note: examine each of the failure classifications – in preparing this list).

Reference
Number

1. Low value of head required by process causing choke or cavitation
2. High value of head required by process causing surge, recirculation or vaporization
3. Fluid density change causing low or high head required by the process
4. Liquid vapor pressure change causing cavitation
5. Foreign object damage from process system (solids, liquids or gas)
6. Improper impeller/seal radial clearances causing items 1 or 2
7. Improper axial clearances causing impeller/stationary passage failure
8. Operating errors – closing suction valve, discharge valve etc.
9. Improper impeller aerodynamic design (head & efficiency produced) causing choke or cavitation or driver overload
10. Improper impeller aerodynamic design (head & efficiency produced) causing surge, recirculation or vaporization
11. Improper impeller mechanical design (stress, bore, seal clearances) causing impeller and/or shaft failure
12. Improper material
13. Improper manufacturing techniques
14. Natural frequencies of rotor and/or impellers excited by operating conditions (critical speeds)
15. Wear out of components due to excessive liquid entrainment

Figure 7.138 Component failure causes – Component: Pump or compressor rotor

Failure cause monitoring parameters

Component: *Pump or compressor rotor*

List all parameters (instruments) to be monitored or used for each failure cause listed in Figure 7.138. (Note: use corresponding failure cause reference number (same reference number as in Figure 7.138).

Reference
Number

1. Process pressures, temperatures, flows and fluid composition
2. Process pressures, temperatures, flows and fluid composition
3. Process pressures, temperatures, and fluid composition
4. Fluid sample, suction pressure and temperature
5. Suction pressure, temperature, flow, fluid conditions and inspection of suction system including suction screen
6. Vibration analysis
7. Vibration analysis, axial displacement and thrust pad temperature
8. Review of process system trend values (machine suction and discharge pressures and flows)
9. Process pressures, temperatures, flows and fluid composition for performance analysis
10. Process pressures, temperatures, flows and fluid composition for performance analysis
11. Vibration analysis
12. Vibration analysis
13. Vibration analysis and process pressures, temperatures, flows and fluid composition for performance analysis
14. Vibration analysis with machine pressures, temperatures and fluid composition
15. Fluid composition, process conditions and vibration analysis

Figure 7.139 Failure cause monitoring parameters - Component: Pump or compressor rotor

2. Anti Friction Journal Bearings

Function Exercise

Component: *Anti-friction journal bearing*

I. 'What' – The function of an Anti-Friction Journal bearing is to support the Rotor in the radial direction.

II. 'How' – The function of the Anti-Friction Journal bearing is accomplished by having sufficient contact area to support the maximum anticipated radial loads, properly installed and being supplied with clean, cool lubricating oil of the proper viscosity to provide the required lubrication for the frictional heat load

Figure 7.140 Function exercise – Component: Anti-Friction Journal Bearing

Component Failure Causes

Component: *Anti-friction journal bearing*

List all possible failure causes. (Note: examine each of the failure classifications in preparing this list).

Reference
Number

1. Insufficient bearing area
2. More load than anticipated
3. Incorrect installation
4. Unclean oil
5. Hot oil
6. Incorrect oil viscosity used or incorrect oil temperature
7. Incorrect oil mass flow rate

Figure 7.141 Component failure causes – Component: Anti-Friction Journal Bearing

Failure cause monitoring parameters

Component: *Anti-friction journal bearing*

List all parameters (instruments) to be monitored or used for each failure cause listed in Figure 7.141. (Note: use corresponding failure cause reference number (same reference number as in Figure 7.141).

Reference
Number

1. Vibration analysis, bearing housing temperature, oil supply and drain temperature
2. Process conditions (flow, fluid composition (SG), suction and discharge pressure, suction temperature), vibration analysis of pipe supports and foundation
3. Vibration analysis, bearing housing temperature, oil supply and drain temperature
4. Oil particle analysis
5. Oil supply temperature
6. Oil analysis and oil temperature
7. Check oil ring movement with strobe light

Figure 7.142 Failure cause monitoring parameters – Component: Anti-friction journal bearing

3. Thrust Bearings

Note: Please refer to the Hydrodynamic Thrust Bearing Example in this chapter, see pages 102–106.

4. Shaft End Seals

Function Exercise

Component: *Pump mechanical seal*

I. 'What' – The function of a Pump mechanical seal is to minimize fluid leakage to the atmosphere

II 'How' the function of a *pump mechanical seal* is accomplished by having the *proper seal pressure – velocity and fluid condition (Vapor pressure, Temperature, Pressure) and seal flush rate*

Figure 7.143 Function Exercise – Component: Pump mechanical seal

Component failure causes

Component: *Pump mechanical seal*

List all possible failure causes. (Note: examine each of the failure classifications in preparing this list).

Reference
Number

1. Improper fluid condition (vapor pressure)
2. Improper pressure in the stuffing box
3. Improper temperature in the stuffing box
4. Plugging of flush line orifice and/or strainer
5. Excessive throat bushing clearance
6. Improper seal assembly
7. Not installing flush line orifice
8. Improper venting of the pump
9. Not chilling down or warming up the pump properly
10. Not allowing tandem seal pots to vent
11. Improper seal balance design
12. Improper seal materials
13. Seal component wearout ('O' rings and secondary seals)

Figure 7.144 Component failure causes – Component: Pump mechanical seal

Failure cause monitoring parameters

Component: *Pump mechanical seal*

List all parameters (instruments) to be monitored or used for each failure cause listed in Figure 7.144. (Note: use corresponding failure cause reference number (same reference number as in Figure 7.144).

Reference
Number

1. Fluid sample
2. Measure pressure in the stuffing box
3. Measure temperature in the stuffing box
4. Measure pressure in the stuffing box
5. Measure pressure in the stuffing box
6. Measure pressure and temperature in the stuffing box
7. Measure pressure in the stuffing box
8. Measure temperature in the stuffing box
9. Measure temperature in the stuffing box
10. Measure pressure in the seal pot
11. Measure pressure and temperature in the stuffing box
12. Measure pressure and temperature in the stuffing box
13. Measure pressure and temperature in the stuffing box

Figure 7.145 Failure cause monitoring parameters – Component: Pump mechanical seal

5. Pressurized lube oil system

Function Exercise

System: *Pressurized lube oil system*

I. 'What' the function of *A pressurized lube oil system* is to *continuously supply cool, clean oil to the components at the required pressure, temperature and flowrate*

II. 'How' the function of *A pressurized lube oil system* is accomplished by *an oil reservoir with interconnecting pipe to main and auxiliary pumps, relief valves, control valves, transfer valves, coolers, filters, pressure transmitters, accumulator and instrumentation to monitor, auto start the pumps and shut down the unit in the event of a malfunction*

Figure 7.146 Function Exercise – System: Pressurized Lube Oil System

Component failure causes

System: *Pressurized oil system*

List all possible failure causes. (Note: examine each of the failure classifications in preparing this list).

Reference
Number

1. Incorrect oil viscosity
2. Incorrect pressure and temperature settings
3. Incorrect cooling water conditions
4. Incorrect component flows
5. Pump suction strainer blockage
6. Pump malfunction
7. Pump driver malfunction
8. Pump coupling failure
9. Relief valve malfunction
10. Control valve instability
11. Control valve excessive friction

12. Accumulator malfunction
13. Transfer valve malfunction
14. Cooler leaks
15. Filter plugging
16. Transfer to a cooler or filter that is not full, vented and heated (in cold climates)
17. Failure to change install filters correctly

Figure 7.147 Component failure causes – System: pressurized oil system

Failure cause monitoring parameters

System: *Pressurized lube oil systems*

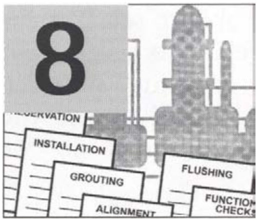
List all parameters (instruments) to be monitored or used for each failure cause listed in Figure 7.147. (Note: use corresponding failure cause reference number (same reference number as in Figure 7.147).

Reference
Number

1. Lube oil sample
2. Check of all pressure and temperature settings
3. Monitor cooling water temperatures
4. Install temporary ultrasonic flow meter to confirm component flows and use control valve position, differential pressure, SG and valve C_v to measure flow where applicable
5. Monitor pump suction strainer differential pressure
6. Monitor pump(s) vibration, seal leakage and bearing temp.
7. Monitor pump driver(s) vibration, seal leakage, bearing temp. and control system (if applicable)
8. Inspect coupling visually and with strobe light during operation if pump and/or driver vibration increases
9. Check relief valves for leaks and proper setting
10. Check control valve controller settings & vent actuator

11. Check control valve actuator diaphragm and packing friction
12. Check accumulator precharge pressure and bladder condition
13. Check transfer valve internals for blockage or excessive friction
14. Check oil sample for water or monitor reservoir level
15. Monitor filter differential pressure
16. Confirm coolers and filters are full, vented and heated (in cold climates)
17. Check oil sample for particle count downstream of filter

Figure 7.148 Failure cause monitoring parameters – System: pressurized lube oil systems



Site reliability assessment

- Site reliability audit form
- Reduction of data
- Identifying targets for improvement

Site reliability audit form

Included at the end of this chapter is a site reliability audit form that I have used successfully throughout my career to obtain input information for reliability assessment work. Readers are encouraged to use the information contained in this document as a guideline when obtaining field input information.

When obtaining input information, it is most important to understand that all input data must be checked for accuracy. If possible, only use recorded DCS trend data and if input information is obtained from site personnel, confirm the accuracy of this information. Accurate facts form the foundation for any site reliability improvement program and will assure that management support will be maintained throughout the program.

Reduction of data

In order to be most effective, the input reliability data from the site audit form must be separated, plotted, inputted into computer analysis, etc. In other words, it must be reduced. In this chapter we will present methods and relationships that have proven to be the most effective ways of analyzing and presenting reliability data. Each approach will first be introduced, defined and an example given. Readers are encouraged to use the information contained in this form to gather site information for reliability assessment work.

At the conclusion of this chapter, the reader will be able to select items that require reliability improvement based on the reliability assessment methods included in this chapter. This information, combined with the information contained in the site reliability audits will form the foundation for reliability improvement recommendations to be presented to management.

Life cycle graph

A life cycle graph, containing years from start-up to the present, plotted on the horizontal (x) axis can be very valuable in determining trends in reliability resulting from operations practices, maintenance practices, condition monitoring parameters, preventive and predictive maintenance procedures. A typical life cycle graph is presented in Figure 8.1 with significant procedural changes noted. Once this graph is developed, any chosen parameter can be plotted on the vertical (y) axis to show the influence of site practices on reliability. Typical (y) axis parameters are:

- Availability
- MTBF
- MTTR
- Cost of unreliability

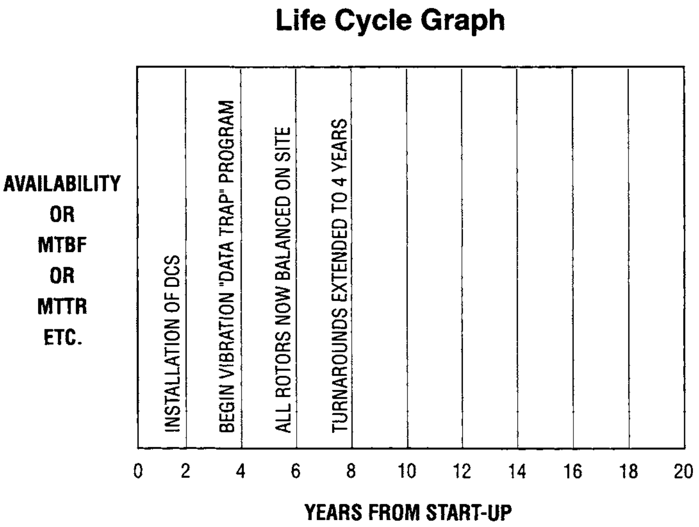


Figure 8.1 Life cycle graph

At the end of this chapter is a typical life cycle graph. Use this graph by noting the significant site changes obtained in sections 1, 3, 4, 5 and 6 of the site reliability audit form. Note these changes by drawing a vertical line and labeling. The vertical line should denote the year the practice started. If the practice was discontinued at a later time, please also note.

Preparation of Pareto Charts

Pareto charts present a clear picture of the major problem areas that reduce reliability. The representation is two dimensional and can plot any number of different parameters. A typical use of pareto charts would be to prepare the following charts from the reliability input data.

A. Number of major replacement parts for your reliability problem.

B. Number of forced outages for each major port.

C. Number of replacement *and* forced outage components for your reliability problem.

An example of a pareto chart for a gas turbine driven compressor unit is shown in Figure 8.2. This could be an example of item C.

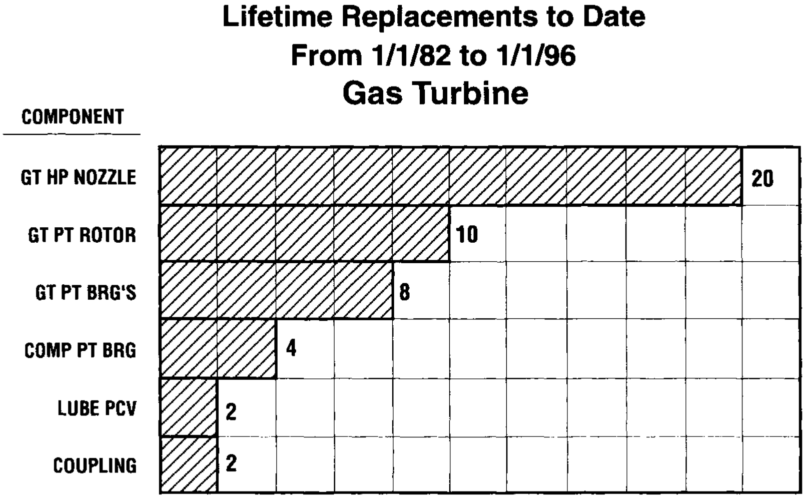


Figure 8.2 A pareto chart

Note the following characteristics:

- Parameters are noted on y axis (could be 'x' axis if required)
- Highest distribution is plotted first

- 'X' scale is direct

Included at the end of this chapter is a pareto chart format for use in assessing replacement components and forced outages.

Determining and measuring availability

Once information regarding failures and repair times is gathered and analyzed, MTBF (mean time between failure), failure rate, MTTR (mean time to repair) and availability can be determined.

MTBF

Mean time between failure is determined by dividing the total operating time for the period to be analyzed by the number of failures in that time period. MTBF can be determined for a unit, a specific piece of equipment or a component. The relationship is noted in Figure 8.3.

MTBF

Mean time between failure

$$\text{MTBF} = \frac{\text{Total operating hours}}{\text{Number of failures}}$$

Figure 8.3 MTBF

I have found that MTBF is most effectively utilized by first assessing data on a machine basis and then analyzed on a component basis (journal bearing, thrust bearing, seal etc) for the major machines ('bad actors') that fail. One should also consider separating MTBFs, machine or component based, into application categories if there is a significant difference between operating parameters (temperature, pressure and/or speed).

As an example, determine the MTBF for an LNG circulating pump given the following data:

1. Operating period 1990 – 1992
2.

Year	*Operating hours*	*Failures*
1990	8600	2
1991	8000	4
1992	8500	1

$$\text{MTBF} = \frac{25100 \text{ Hours}}{7 \text{ Failures}}$$

MTBF = 3586 hours

Failure rate

Failure rate is the number of failures per machine hour. In other words, it is the reciprocal of MTBF. Figure 8.4 presents failure rate.

Failure rate

The number of failures per machine hour

or failure rate = $\frac{1}{\text{MTBF}}$

Figure 8.4 Failure rate

For the example on page 248, the failure rate for the LNG circulating pump is:

$$\text{F.R.} = \frac{1}{3586} = 2.789 \times 10^{-4} \text{ per hour}$$

MTTR

Mean time to repair is the total time to repair a unit, equipment item or component during a specific time period divided by the number of repairs.

MTTR

Mean time to repair

MTTR = Total number of repair hours for a specific

- Unit
- Equipment item
- Component

Divided by number of repairs

Figure 8.5 MTTR

As an example, determine the MTTR for a gas turbine during a 24 month period as noted below.

Repair	Date	Repair description	Total hrs*
1	1/1/93	Replace H.P.T. nozzles	96
2	4/8/93	Replace fuel nozzles	72
3	7/20/93	Replace No. 1 & 2 bearings	30
4	12/23/93	Replace P.T. rotor and bearings	36
5	5/6/94	Replace H.P.T. nozzles	80
6	11/15/94	Replace compressor (L.P.) rotor, stators and bearings	80

*Includes cool down time

Total maintenance hours =	394
Number of repairs =	6

$$\text{MTTR} = \frac{394 \text{ hours}}{6 \text{ repairs}}$$

=65.67 hours

Availability

Availability is a more effective measurement of reliability since availability is the percentage of time that a unit or equipment item operates compared to the time it is available to operate. Like reliability, it is normally used as a measurement for critical (unspared) equipment. Availability can be directly expressed as a function of time or as a function of MTBF and MTTR as shown in Figure 8.6.

Availability

$$\text{Availability} = \frac{\text{No. of operating hours/year}}{8760 - \text{planned downtimes (T\&I's or turnarounds)}}$$

$$\text{Availability} = \frac{\text{MTBF}}{\text{MTBF} + \text{MTTR}}$$

Figure 8.6 Availability

As an example, if the MTBF for the gas turbine in the previous example is 2836 hours, what is this G.T.'s availability?

Given: MTBF = 2836 hours
MTTR = 65.67 hours

$$\text{Availability} = \frac{2836 \text{ hours}}{2836 + 65.67 \text{ hours}}$$

$$= 97.73\%$$

Included at the end of this chapter, is an availability factor worksheet that can be used to determine unit or component MTBF, failure rate, MTTR and availability.

Select the unit with the lowest reliability in your plant over the past two years and plot availability and MTTR for each year since the process unit startup on a life cycle graph.

Identifying targets for improvement

Once the site reliability audit data has been reduced, areas for reliability can be identified. In the previous section, the areas with the lowest availability were identified progressively as shown in Figure 8.7.

Identifying targets for reliability improvement

Train availability

↓

Unit availability

↓

Item availability

↓

Component availability

Figure 8.7 Identifying targets for reliability improvement

Normal component reliability comparison

Once low availability machinery items or components are identified, they must be compared to normal values to determine if a reliability improvement program is warranted. Normal availability data is available from company data bases, data obtained during Industry conferences and from personal experience. A suggested source for comparison is Table 4.3 from Machinery Reliability Assessment by Heinz P. Block and Fred K. Geitner, © 1990 by VanNortrand Reinhold. This table contains 'best' and 'worst' failure rates for a variety of components as well as basic failure modes. At this point, it is suggested that you list the component failure rates for your plant's lowest availability equipment item. Then compare the site actual to the normal failure rates and determine if corrective action is required. A component reliability comparison worksheet is included at the end of this chapter.

Cost of unreliability

At this point, the 'bad actors' or the 'hit list' has been identified and the specific availability measurements quantified. What remains is perhaps the most difficult task. See Figure 8.8.

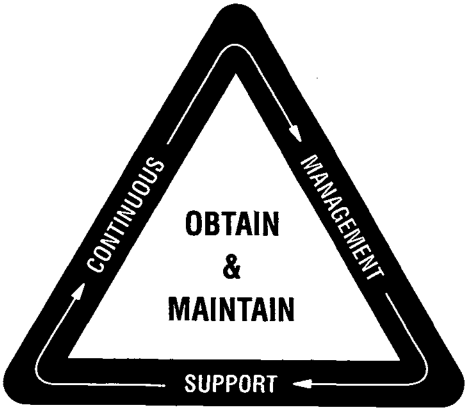


Figure 8.8 The most difficult task

Regardless of how great your salesmanship is, you will not succeed unless your plan is 'cost effective' in management's opinion.

Therefore, a 'cost' must be assessed for each bad actor. We define this cost as the **'cost of unreliability'**.

The cost factors are stated in Figure 8.9.

The cost of unreliability critical rotating equipment (per year)

- Lost product revenue x *days forced outage*
- Maintenance costs
- Replacement part cost
- Labor cost
- Unnecessary turnaround time*

*Assumes process unit start-up is delayed by activity

Figure 8.9 The cost of unreliability critical rotating equipment (per year)

Simply add the costs of unreliability for each component that does not meet the component reliability norms. Once these figures are obtained, the reliability assessment process is complete. We are now equipped with the data to proceed up the reliability pyramid to prepare reliability improvement plans. At this point, use the cost of reliability worksheet, included at the end of this chapter to tabulate the costs of unreliability for the components of your 'bad actor' list.

This completes the chapter on reliability assessment. The next chapter will address the preparation of a site reliability optimization plan.

Site rotating equipment reliability audit

Process unit: ____________________

Included items: ____________________

Study team member	**Work discipline**
1. ____________	____________
2. ____________	____________
3. ____________	____________
4. ____________	____________
5. ____________	____________
6. ____________	____________

It is recommended that the study team consist of the following disciplines as a minimum:

- Reliability
- Operations
- Process engineering
- Maintenance
- Control and instrumentation
- DCS specialist

I. General information

1. Daily revenue loss (local currency)
   A. If critical equipment (unspared) experiences a shutdown

   ____

   B Daily process unit production (tons/day) ____________
2. Contractor data
   A. Engineering contractor ____________
   B. Construction contractor ____________
   C. Start-up date ____________
3. Operations data
   A. Control room modernization

1. Changes and dates of change ______________________

____

____

B. Operator condition monitoring responsibility

Example	Responsibility	Date started
	Overall pump vibration (monthly)	1/1/90
	______________________	__________
	______________________	__________

4. Maintenance data

A. Proactive maintenance

1. Do maintenance personnel become involved with determination of root cause problems? Yes or No

B. Site maintenance

1. Since train start-up, have there been changes concerning the maintenance performed on site. Yes or No
2. If yes, what changes and when.

Example	Dates	Changes
	______________________	__________
	Started to balance on site ______________	1/1/94
	______________________	__________
	______________________	__________

C. Hourly cost of maintenance labor ______________________

5. Condition monitoring data

A. Method (I.E. manual vs. 'data trap' vs. D.C.S.)

Example	Method	Years used
	Manual logs	1981-1988
	______________________	__________
	______________________	__________
	______________________	__________

B. Condition monitoring parameters. Please answer if the following parameters are condition monitored (see note)

1. Centrifugal compressor performance parameters (P_1, T_1, P_2, T_2, Flow, Speed, Gas Analysis) Yes No
2. Pump performance parameters (P_1, P_2, S.G., Flow & Speed) Yes No
3. Compressor seal fluid (gas or oil) supply flows (or valve position if oil system) (Gas) Yes No
   (Oil) Yes No
4. Centrifugal pump seal stuffing box pressures and temperatures Yes No
5. Centrifugal compressor balance line ΔP Yes No

Note: In the space provided between items, state item numbers of units where the answer is 'Yes'.

6. Preventive maintenance data

A. Please answer if the following items are changed on a preventive (time interval) basis. If the answer is Yes, please state the time between replacement time.

	Time		
1. Compressor rotors	_____	Yes	No
2. Compressor bearings	_____	Yes	No
3. Compressor seals	_____	Yes	No
4. Compressor labyrinths	_____	Yes	No
5. Gas turbine H.P. rotor	_____	Yes	No
6. Pump bearings	_____	Yes	No
7. Pump seals	_____	Yes	No
8. Pump wear rings	_____	Yes	No
9. Electrical switches	_____	Yes	No
10. Accumulator bladders	_____	Yes	No
11. Control valve diaphragms	_____	Yes	No
12. Oil filter cartridges	_____	Yes	No

B. Please answer if the following items are cleaned on a preventive basis.

	Time		
1. Oil reservoirs	____	Yes	No
2. Cooler tubes	____	Yes	No
3. Control valve pulsation dampeners	____	Yes	No
4. Seal oil drain traps	____	Yes	No
5. Seal oil degassing tanks	____	Yes	No

II. Turnaround activities

Please complete the following form for each noted equipment unit (this includes driver, driven, transmission device and auxiliary systems)

Plant ____________________
Process unit ________________
Compressors

Type and item no.	Turn-around dates	Major item replaced*	Reason†	Part cost	Labor time (hours)

* Major items: Rotors, coupling, bearings, seals, control valves
† Use following code
P.M. = Preventive maintenance
P.D.M. = Predictive maintenance indicated imminent failure
F.P. = Failed part

Plant ____________________
Process unit ________________
Pumps

Type and item no.	Turn-around dates	Major item replaced*	Reason†	Part cost	Labor time (hours)

* Major items: Rotors, coupling, bearings, seals, control valves
† Use following code
P.M. = Preventive maintenance
P.D.M. = Predictive maintenance indicated imminent failure
F.P. = Failed part

Plant ____________________
Process unit ________________

Equip. type and item no.	Date of event	Reason for outage	Parts affected	Process unit outage hrs.	Time to repair (hours)

Cost of parts	Suspected causes	Corrective action

III. Major rotating equipment forced outages

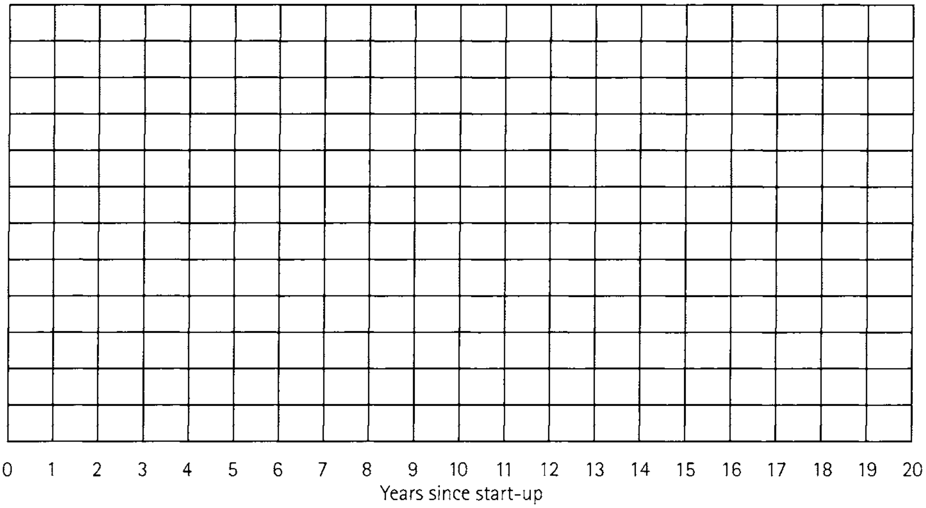
Please note the forced outages of major equipment, along with each item n umber for your group's assigned process unit.

IV. Rotating equipment documentation

Based on major forced outage events (occurring more than once) in section III, obtain the appropriate equipment data sheets, performance curves (if applicable), assembly drawings and P & ID's for these items.

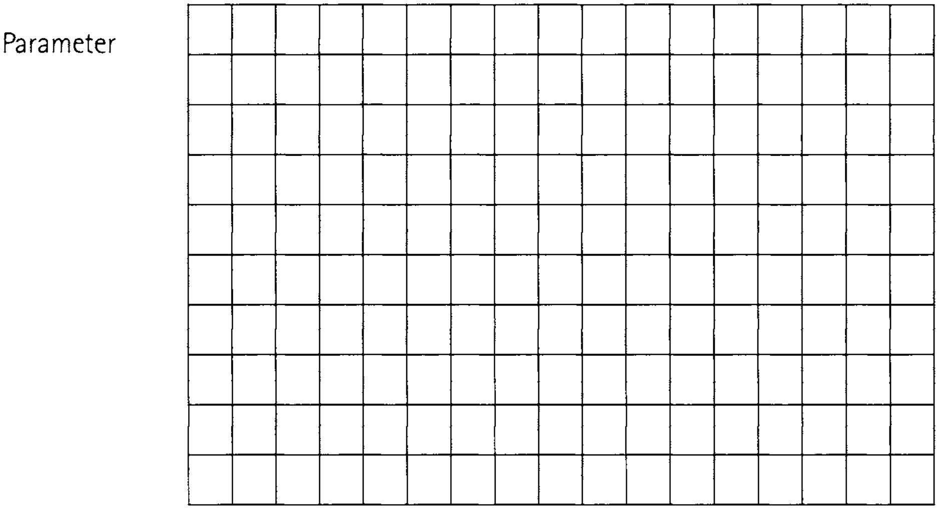
Life cycle graph

Date ________ Group ________ Train ________



Pareto chart

Chart title ____________________ Group ________________________
Time period ____________________ Train ________________________
Item ____________________ X Scale: 1 block = ___________items



Availability factor worksheet

Date ________ Group ________ Train

Unit/component	MTBF	Failure rate	MTTR	Availability

Component reliability comparison worksheet

Date ________ Group ________ Item ________

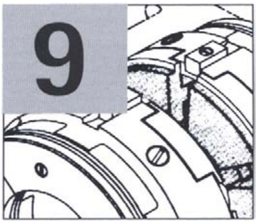
Component	Failure rate	Normal failure rate		Action required	
		Best	Worst	Yes	No

Cost of unreliability worksheet

Date ________ Group ________ Item ________

Component	Costs			
Labor	Material	Revenue	Unnecessary turnaround*	Total

*Unnecessary turnaround (ta) time assumes start-up is delayed by activity



Preparing a site reliability optimization plan

- Introduction
- Identifying opportunities for optimization
- Determine the root cause of each identified opportunity
- Establish steps to prevent re-occurrence of problems
- Setting up an effective multi disciplined site reliability initiative
- Obtain and maintain management support
- How to maintain continuous improvement of the established program
- Appendix
  - Questionnaires
  - Design Audit Example 1 Rotor axial (thrust) forces tutorial
  - Design Audit Example 2 Lube system design tutorial
  - Management awareness workshop agenda example

Introduction

Regardless of the level of a site reliability optimization program, it can be improved. My personal experience is that:

- Bad actors* are defined but not permanently solved
- The majority of maintenance activities still are reactive – 'Firefighting'
- Preventative maintenance (time based) activities are excessive
- Predictive maintenance (condition based) activities still are minimal
- Some action plans for bad actor resolution have been defined but are not implemented
- Lessons learned, from bad actors, have not become best practices for machinery upgrades and/or new equipment purchases.
- The reliability effort is maintenance based and does not incorporate other site disciplines.

*A bad actor is defined as any machine or stationary item that experiences one or more ESD's per year (un-scheduled shutdowns or failures). Note that while this book is devoted to rotating equipment, the principles discussed in this chapter are equally applicable to all the equipment in any plant.

Why are these characteristics prevalent in most plants? My opinion is that the plant management has not been convinced of the opportunities available to save considerable operating costs and increase profit by reducing downtime related to reliability issues and excessive maintenance time. Plant management must be shown results and the associated savings to endorse any plan for reliability optimization. Often, a 'salesman' that has the ear of management is required. I have found that this person is usually a trusted, senior operator or a process engineer. And considering that most reliability efforts are maintenance based, this salesman is nowhere to be found.

This chapter will therefore cover all the steps, in clear concise terms from identifying the problems and turning them into opportunities, thru the development of a reliability program and the continuous improvement of that effort to make it an established world class site reliability optimization program.

Identifying opportunities for optimization

Begin by using all available site information and process this information to determine the 'targets' that present the greatest opportunity for increased profit. The site reliability audit form contained in Chapter 8, and its associated availability worksheets are a guideline and starting point for obtaining the input information. Once this information is obtained the MTBF or failure rate, MTTR, availability and cost of un-reliability for the most costly 'bad actors' on site can be identified as opportunities for optimization. The 'targets' should also consider the time and cost of the resolution effort. These issues are noted in Figure 9.1 below.

Target opportunity guidelines

Select the bad actors with:

- The lowest MTBF
- The highest cost on un-availability

That are:

- The easiest to resolve
- For the lowest cost to resolve

Figure 9.1 Target opportunity guidelines

I have found that once the input information is obtained (Chapter 8 Audit form or other means), further questionnaires issued separately to Maintenance, Operations and Engineering provide additional valuable information to define key issues concerning the culture of the plant and possible root causes based on the procedures used. These questionnaires are contained in the back of this chapter. Separate questionnaires are supplied for Maintenance and Maintenance Engineering and Operations and Process Engineering. This information can be either emailed to the appropriate people or used as a checklist if questions are asked personally.

I have used these forms successfully prior to visiting a plant and usually have cycled the questionnaires 2 or 3 times to enable me to obtain as much information as possible prior to entering the plant and to prepare a site audit agenda that optimizes the effectiveness of the time on site and minimizes the time necessary for interviews. Figure 9.2 presents the form of a typical site audit agenda we have used.

Site audit agenda – example

Day 1 – am	■ Assemble audit team – introduction ■ Review completed audit questionnaires ■ Confirm schedule for each day of audit
pm	■ Visit first process unit ■ Daily review of results
Day 2 – am	■ Review day 1 activity report ■ Define remaining activity for first process unit
pm	■ Visit site for remaining first process unit activity ■ Daily review of results
Day 3 – etc.	■ Same activity as day 2 for each process unit
Last day – am	■ Management presentation

- Present
  - Audit procedure
  - Observations
  - Recommendations
- Emphasize that intent is to 'test' all observations and recommendations prior to final report

Figure 9.2 Site audit agenda – example

The audit agenda will also define the makeup of the individuals that need to participate. It is advisable to assemble a site audit team of experienced personnel prior to the visit. This action will also aid in having the required information assembled or even sent to the auditor prior to the site visit since the assigned team members can perform this task. Figure 9.3 presents the disciplines typically assembled for an effective site audit team.

Suggested site audit team make-up

One experienced member should be on the team from each of the following disciplines:

- Operations
- Maintenance
- Process engineering
- Reliability engineering
- Maintenance engineering
- Instrumentation and control

Figure 9.3 Suggested site audit team make-up

At the conclusion of the site audit and/or completion of the questionnaires, all the required input information will be available to identify the site opportunities for reliability optimization. An example of a proposed site opportunity list is shown in Figure 9.4.

Site reliability optimization opportunity list example

- Site awareness training
- The 5 most costly bad actors
- Increase PDM activities in 2 plant areas
- Minimize plant PM activities in 2 areas
- Optimization of spare part storage and retrieval

Figure 9.4 Site reliability optimization opportunity list example

Note that this opportunity list not only includes bad actors but site procedure change opportunities such as increased PDM, reduced PM and spare part procedures. When preparing this list be sure to benchmark all recommendations compared to other plants in your company or the industry. Be honest, the lower the ranking of your unit, the better the opportunity for improvement – if you don't lose your job in the process!!

In my experience, I have led many a group on a successful benchmarking trip that has resulted in not only valuable information for us but also the visited plant and has often resulted in the establishment of a valuable user network. Figure 9.5 presents an example of benchmarking recommendations.

Benchmark all opportunities

Opportunity	Benchmark	
Awareness training	The company Houston	plant
Increase PDM	Shell ______________	plant
Form seal taskforce	Exxonmobil ___________	plant

Figure 9.5 Benchmark all opportunities

It is very important to have benchmarks to demonstrate to management the need for improvement and/or that the proposed recommendation has been successfully implemented in an effective life cycle cost manner at other industry locations. It is most effective if benchmarks can be used from similar plant processes in the same geographical region. Figure 9.6 demonstrates the effectiveness of this approach.

Benchmarking guidelines

Effective references (benchmarking) of successful problem resolution have the following characteristics:

- Taken from similar process units
- Taken from your company if possible
- In the same country
- Visit approved, with final agenda, 2 weeks prior (usually with reciprocal visit arranged)

Figure 9.6 Benchmarking guidelines

Be sure to list the incurred cost of un-availability for each major bad actor as well as procedure improvement recommendations. Refer back to Chapter 8 for calculation of the cost of un-availability and note that the most important part of this calculation is the cost of incurred process unit downtime or rate reduction. This is the 'hammer' that will 'nail' the recommendation. Refer to Figure 9.7 regarding a suggested management presentation format.

Obtain and maintain management support by ...

1. Clearly stating impact of problem on plant profit (cost of unavailability)
2. Prepare a brief statement of:
    - Problem
    - Impact on plant
    - Reliability improvement plan
3. Be confident!
4. Be professional!
5. Be autonomous (do not expect management to do your job!)
6. Provide timely updates

Figure 9.7 Obtain and maintain management support by ...

Also do not forget to 'walk before your run'. Do not attempt to incorporate every opportunity into the program at this point, you want to prove the profitability of the program first, there will be plenty of time to add items. Choose the 'big bang for the buck' targets (lowest cost for the highest return) first.

Following a successful management presentation, you should have the required management support budget and necessary resources to proceed to the next step of the program. See the management support pyramid in Figure 9.8.

The most difficult task

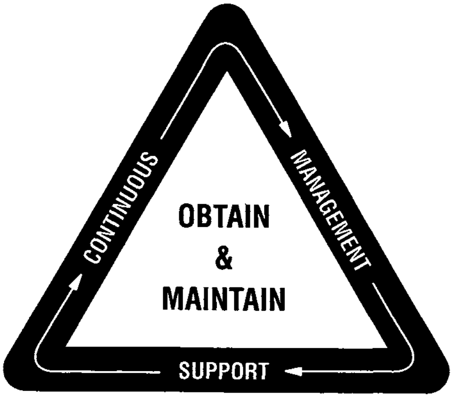


Figure 9.8 The most difficult task

A detailed report of the audit is both expected and required to serve as a checklist for the program status and success. It is very important to list all salient points pertaining to observations and recommendations in the management power point presentation (PPT) prior to leaving the site so that the validity of all facts can be confirmed and so that there will be 'no surprises' in the final report. Figure 9.9 presents a typical management outline format for your use in preparing a management 'asset opportunity improvement' presentation.

Presentation outline

- Study objective and methodology
- Presentation objective
- Study observations
- Study recommendations

Figure 9.9. Presentation outline

To complete this section, I have noted the following salient points that I have experienced during the audits that I have performed over the years:

- The most frequent machinery bad actor types are: pumps, general purpose turbines and auxiliary systems.
- The most frequent application areas for bad actors are: pumps in light and heavy hydrocarbon service, general purpose turbines in low pressure steam applications, transient events on lube/seal oil systems and dirt entrainment in dry gas seal systems.
- The most frequent component problems: pump seals and bearings, general purpose turbine carbon ring seals and governor linkages, dry gas seals (face seal blockage and secondary seal hang up) and auxiliary system oil systems components – control valves and small turbine linkages.
- The standard industry approach concerning bad actors is to group only according to equipment type and not to group according to application and components. Grouping according to application and components allows greater focus to determine the specific root causes.
- My experience with the bad actor approach and its subsequent root cause analysis procedure is that the problems are solved only for the short term and not for the long term and repeat over time.

- 80% or greater of the actual root causes of failure lie in the effects of the process or support systems on the equipment components.
- Management does not totally commit to the implementation of the proposed action plan.
- Continuation of a successful program must consider upper management change and the need to demonstrate credibility to each new regime.

Determine the root cause of each identified opportunity

Once the targets are identified, use the RCA procedure in Chapter 5 to determine the root cause of each identified opportunity. Do not forget to examine all 5 of the failure classifications when looking for causes and root causes. Start with the technical issues but remember that the actual root cause of many of the technical issues may in fact be an inadequate site procedure or lack of one! An example would be the continuous mechanical seal failures for a light hydrocarbon application. Refer to Figure 9.10.

Procedural root cause example

- Determined root cause = low seal chamber pressure causing premature vaporization across seal faces resulting in 5 or more failures per year
- Actual root cause = lack of an established site seal system condition monitoring program (refer to material in Chapter 4 for parameters to monitor)

Figure 9.10 Procedural root cause example

Once the target opportunity root cause is identified be sure to develop an easy to implement, cost effective action plan. Make sure that the solution is quick to implement and low cost. Refer to Figure 9.11 for the action plan generated for the *example* in Figure 9.10.

Mechanical seal problem resolution action plan

- Drill and tap affected pump seal chamber for installation of a pressure gauge
- Use available pressure gauges in the site warehouse
- Establish an operation procedure to monitor pressure gauges and define acceptable limits.
- Conduct mini-info awareness sessions to obtain implementation and ownership among operators

Figure 9.11 Mechanical seal problem resolution action plan

Many times the defined root causes point to a failure classification that requires contractor, vendor or consultant input. In these cases, a question list should be constructed and sent to the appropriate party well in advance of any discussion or meeting. An example of a vendor information request form is shown in Figure 9.12. Note that this form could also serve as a meeting agenda if required.

General purpose steam turbines reliability improvement issues

Turbine plant item numbers

____

____

Turbine vendor serial numbers

____

____

I Reliability improvement issues

1. Installation of independent electronic over-speed devices
   We wish to install electronic over-speed devices on the above noted turbines. Please advise the following:
   a. Modifications required for each turbine
   b. If the present mechanical over-speed device can be disabled by tightening the adjustment screw. We want to eliminate the mechanical device, but we do not wish to rebalance the rotor. Therefore, we would prefer securing the present over-speed device (over-speed bolt).

2. Installation of a vacuum device in the steam leak off ports to positively prevent steam leakage from entering the bearing housing.

   We wish to modify the following turbine item numbers: ______, _____

   The item numbers are turbine serial numbers: ______, _____

   ______________, and ______________

   Please provide the following information:

   a. Approximate steam and air leakage for each end seal.
   b. A proposal for steam eductors. The steam conditions available are:

      P___________

      T =___________

   c. The approximate steam flow required for each eductor.

Figure 9.12 General purpose steam turbines reliability improvement issues

Following an initial discussion with the vendor, a design audit may be required to determine the actual root cause. We have included two examples in this chapter of audit formats required to resolve long term, costly field reliability issues – balance drum and lube oil system audit in the back of this chapter. Additional information regarding design audits is contained in Chapter 10.

If the defined root cause is a procedure issue, contact with the vendor and/or the process licensor may be required. Figure 9.13 presents an example of such a case.

Compressor liquid bushing seal repeated failure RCA example

- Defined root cause – gas pressure applied to the compressor before the liquid seal system was put into operation causing debris to enter and plug the small clearance (0.002–0.004) between the inner seal bushing and shaft sleeve.
- Action – review of instruction book and confirmation discussion with vendor to revise site operating procedure and training with operations to absolutely assure that the seal oil system is always put into operation before gas is introduced into the compressor case.

Figure 9.13 Compressor liquid bushing seal repeated failure RCA example

In Figure 9.14, I have presented a typical design audit meeting agenda to resolve a long term compressor seal oil system problem. Always remember to be sure to send agendas well in advance to allow the vendor sufficient time for preparing the required material.

Design audit agenda

Lube/seal oil system

1. Introductions
2. Purpose of meeting
   2.1 Review study results, past modifications/failures and recommendations to assist client in resolving seal oil delta pressure trips on the subject compressor.
   2.1.1 Supply client with recommendations, modifications required and cost and delivery.
3. Results of studies performed in the field
   3.1 Client field reliability study
   3.2 Vendor engineering study
4. Review seal design and requirements
   4.1 Review seal components and function
   4.1.1 Upgrades?
   4.1.2 Modifications?
5. Review seal oil system component design
   5.1 Sizing of components (pumps, coolers, filters, reservoir, etc)
   5.2 Review valve selection and sizing (including Cv)
   5.3 Review control system
   5.3.1 Upgrades?
   5.3.2 Modifications?
6. Review comments to seal oil system component sizing study
   6.1 List recommendations
   6.1.1 Feasibility and reliability issues
7. Review final recommendations and feasibility
   7.1 Assign tasks and schedules
   7.2 Create final timeline up to delivery of parts and installation
8. Conclusion

Figure 9.14 Design audit agenda

In conclusion, be sure to minute all meetings and/or conference calls in detail and to define action and the responsible personnel. Guidelines are presented in Figure 9.15.

Always record all details

- Accurate, approved minutes
- List all major details in simple, concise terms
- Define each reason for proposed action
- Define required date and responsible individual
- Follow-up until all action items are completed

Figure 9.15 Always record all details

Establish steps to prevent re-occurrence of problems

I cannot tell you how many times I have solved the same problem ... in the same plant. Does this scenario sound familiar? Why is it the rule rather than the exception? In my opinion it has to do with the state of asset function awareness on site. Wherever I have worked, the state of the awareness of the design function of the installed assets can be improved. True, all plants have experienced engineering, operations and maintenance personnel. However, they rarely have had the opportunity to be directly involved with the design of installed equipment. I have found that most plant personnel crave for this knowledge regardless of their experience level. Once the experienced people on site become more aware of asset function, they naturally acquire a spirit of ownership in terms of condition monitoring and problem analysis. This leads directly to a site culture change and the beginning of a predictive maintenance approach. I have encouraged our clients to adopt the philosophy shown in Figure 9.16.

Reliability is everyone's responsibility

Regardless of position or discipline!!!

Figure 9.16 Reliability is everyone's responsibility

How can component function awareness be acquired? Is an outside consultant required? There are many alternatives depending on your company culture, budget and staffing. I have found that the most effective approach is to initially use a corporate specialist, supported by the corporate training department or an experienced outside consulting firm (If your 'corporate approach' is no longer in place) to present the training to site experienced, interpersonal communicators who in turn will train the remaining staff. Some approaches are presented in Figure 9.17.

Awareness improvement on site

- Mini-information sessions presented by site personnel
- Newsletters and/or emails relating the cause and solution of site reliability issues
- Asset function training presented by in-house specialists
- Asset function training presented by corporate specialists
- Asset function training presented by vendor specialists
- Asset function training presented by outside consultants

Figure 9.17 Awareness improvement on site

What should the objective of the training, regardless of the approach be? The objective is to explain what the function of a specific asset and its major components are and how they accomplish the stated function. The presentation should be brief and specific with a specific site example or two noted. And the training should include a group visit to the actual asset requiring the group to perform an exercise to assure implementation of the principle. Figure 9.18 presents these important facts.

Suggested format of site awareness training
'Talk – Walk – Talk'

- Include all disciplines (operations, maintenance & engineering) in the training group
- Define asset and major component functions
- Use actual site drawings and documents
- Define how the functions are performed
- List the parameters to be monitored
- Visit the asset and perform group condition monitoring exercises
- Return to the classroom, discuss results and formulate action plans

Figure 9.18 Suggested format of site awareness training

An important consideration is to be sure to integrate the training group using all affected site disciplines. I have found that this approach definitely improves site overall awareness and a spirit of teamwork that significantly reduces 'finger pointing'.

Before leaving the subject of site awareness training, one additional question has to be asked ... Who receives the training? Answer ... Everyone!!!

A 'top down' approach is required. If the personnel responsible to approve action plans are not aware of the consequences, in terms of lost profit, of not approving recommended action, the plan will never be implemented! I have taken the approach to separate management training from the other site training and usually use a corporate specialist or outside consultant company for this purpose.

It is suggested that consideration also be given to retain the same individual or company to provide the first session of component function awareness training to site senior people who are respected on site for their experience and ability to effectively communicate. I call these people the 'believeables'. An agenda for a 2 day management awareness workshop is presented in the back of this chapter.

Once site awareness is on the rise, it will be time to start to 'condition everyone' literally!!! The next step is to introduce the concept of CCM (Component Condition Monitoring) presented in Chapter 4. As you will remember, CCM requires that each of the 5 major components in any type of rotating equipment be monitored using all of the required parameters for each component. Figure 9.19 presents the five (5) major components in each piece of rotating equipment.

The five major rotating equipment components

- Rotor
- Journal bearing
- Thrust bearing
- Seals
- Auxiliary systems

Figure 9.19 The five major rotating equipment components

While this book is written with rotating equipment in mind, the principles presented can also be applied to stationary equipment.

How can CCM become part of the plant culture? I suggest that an example of CCM be included either in a mini information session or that a 'Talk – Walk – Talk' session be presented as detailed in Figure 9.18.

Try to minimize the length of the training sessions especially if your plant has not done this in the past. But be sure to start implementing the principle as soon as possible and include maintenance and operations in the program. In newer plants, it is easy to incorporate CCM into an Excel Format that can provide trends of all desired parameters using the site process information system (PHD, PI, ASPEN etc.). Figure 9.20 presents an example of the parameters to monitor for a bearing.

Condition monitoring parameters and their alarm limits

Journal bearing (anti-friction)

Parameter	Limits
1. Bearing housing vibration (peak)	.4 inch/sec (10 mm/sec)
2. Bearing housing temperature	185°F (85°C)
3. Lube oil viscosity	off spec 50%
4. Lube oil particle size	■ Non metallic = 25 microns
5. Lube oil water content	■ Metallic - any magnetic particle in the sump below 200 PPM

Figure 9.20 Condition monitoring parameters and their alarm limits

Remember that CCM is the basis of predictive maintenance (PDM) but this concept must be practiced site wide and requires implementation, practice and acceptance by all site disciplines.

A good example of 'conditioning everyone' into a new site culture is the refinery that had persistent reliability issues in a certain area of the plant. Using awareness training of operations, maintenance and engineering by dedicated site personnel, the concept of the equipment reliability operating envelope (EROE) was implemented. This approach uses the CCM for a Rotor, see Figure 9.21 below, to determine the operating point of each pump.

Pump performance monitoring 'the rotor'

1. Use calibrated instruments
2. Measure:
    - P_1
    - P_2
    - Flow
    - Specific gravity
3. Calculate:

    Head produced $\frac{FT\text{-}LB_F}{LBM} = \frac{\Delta P \times 2.311}{S.G.}$
4. Determine flow rate from either:
    - Flow meter
    - Control valve
    - Motor amps
    - Steam turbine throttle valve position
5. Plot the operating point on the pump shop test curve to assure the operating point is in the equipment reliability operating envelope (EROE)*
6. Take appropriate action if required

*EROE is typically -50% to +10% in flow of the pump best efficiency point (BEP)

Figure 9.21 Pump performance monitoring 'the rotor'

This approach was used since all of the affected pumps suffered from the effects of the process causing the pump to operate outside the heart of the pump curve, see Figure 9.22, thus being the root cause of all the damage suffered by these pumps (repeated bearing, seal and wear ring failure).

Operations saw the value of this approach and immediately implemented a program using local monitoring to maintain the operating point in the heart of the curve and remove these pumps from the 'bad actor' list ... for now! The challenge will be to see if this program can be maintained!

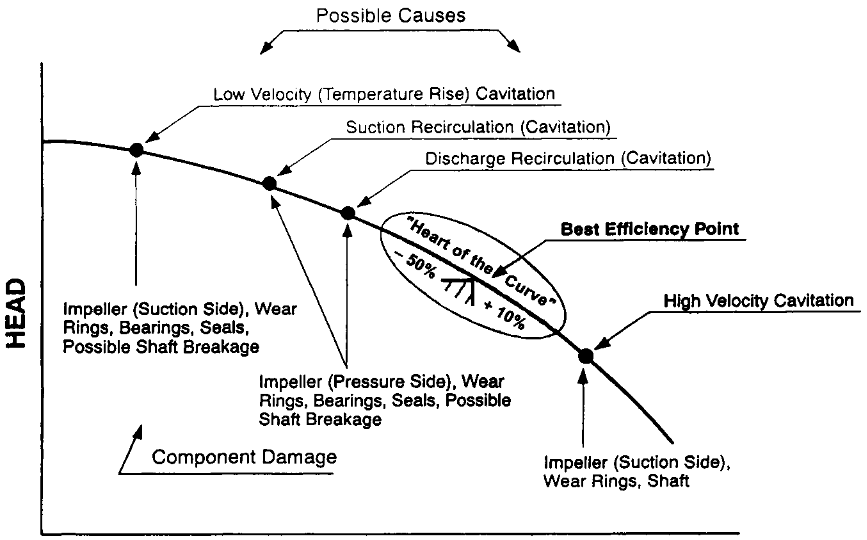

Possible Causes
Low Velocity (Temperature Rise) Cavitation
Suction Recirculation (Cavitation)
Discharge Recirculation (Cavitation)
"Heart of the Curve"
- 50%
+ 10%
Best Efficiency Point
High Velocity Cavitation
HEAD
Impeller (Suction Side), Wear Rings, Bearings, Seals, Possible Shaft Breakage
Impeller (Pressure Side), Wear Rings, Bearings, Seals, Possible Shaft Breakage
Component Damage
Impeller (Suction Side), Wear Rings, Shaft

Figure 9.22 Centrifugal pump component damage and causes as a function of operating point

Along these same lines, I was recently involved with a new oil field project where all critical pumps were designed with pressure and flow transmitters to automatically assure that pumps were controlled to stay in the heart of the curve. In the control room, the 'Live' operating point was displayed on the shop test pump differential head vs. flow curve to allow operations to see if the pump was operating in the 'Heart of the Curve' and if the pump performance had deteriorated (if the operating point was not on the shop test, new condition, curve).

Once the concept of PDM starts to take hold, it is time to begin to whittle away the expensive and extensive PM (preventive maintenance – time based) list! Optimize your existing PM program based on the results of the PDM program. Figure 9.23 presents these facts.

Optimizing PM based on PDM results

- Use CCM monitoring parameters to determine condition of components prior to and after the established PM interval
- Extend the interval based on a constant (no significant change) trend of the affected parameter
- An example is extending the established 26 week pump oil change interval to an interval of 39 weeks, and eventually 52 weeks, based on using the CCM principle of monitoring bearing parameters (see Figure 9.20)

Figure 9.23 Optimizing PM based on PDM results

Finally, remember that any significant site culture change will mean extra work initially and perhaps involve additional personnel. Do not fall into the trap that I have and be sure to focus on a few ‘big bang for the buck’ (targets that are costly, high exposure ones) to prove the viability of the program to management and therefore assure improvement and continuation of the program. Also do not forget the most important part … ‘Reliability is everyone’s responsibility’ and be sure to include multi-disciplines in the effort. And consider helping the future engineers of your country and use students (Interns, Co-op’s … whatever you call them). It is truly amazing that young, dedicated, bright students cannot obtain needed training during their engineering studies. The important points to consider are noted in Figure 9.24.

Easing the transition

- Select the targets that can show immediate cost savings
- Do not overload the staff
- Assure that the effort includes all site disciplines
- Use engineering students to assist where possible
- Report results effectively to management

Figure 9.24 Easing the transition

Do not forget that continued improvement of the knowledge base of asset component function is essential to the success of the program. Refer to the principles presented in Chapter 7 for guidance and get the message across to all site participants.

Setting up an effective multi disciplined site reliability initiative

In my experience, I have worked with many site reliability efforts, different names, different cultures and different levels of experience. In the early days of my experience with these efforts (1970’s and 1980’s) there was but one constant … all efforts were strictly maintenance based. Oh, yes, there was another constant … identified opportunities for reliability improvement were not implemented.

The efforts also usually had a name and some of those were: the reliability group, the vibration monitoring group, the failure analysis unit, the failure analysis and PM group etc., etc. Regardless of the name, the effort was aimed at improving site reliability of all equipment, not only rotating equipment, and definitely achieved results but they were usually short term and problems re-occurred. Over the years, as is natural, the effectiveness of the program in a certain plant varied as personnel entered and left the group. Those who left, usually did so for higher pay and those that entered did so for increased experience.

Review of the historical efforts of these groups also had an interesting similarity. This was that every group or plant effort usually had identified the root cause of a particular problem but was unable to gather sufficient continued management support to implement the identified action plan. As a corporate consultant for a major oil and chemical company and later as an independent consultant, I would be asked to review various site reliability problems and recommend a cost effective action plan. In the majority of the cases, my end result was very close to what had been recommended prior to my study. However the difference was that my action plan was usually accepted and worked (just as the site group's plan would have also been successful!). This is not very encouraging to someone who works long hours, is 'on call' and has to attend to reactive maintenance issues at the usual times ... weekends!!! Incidentally, have you ever noticed that equipment usually decides to 'pack it in' on the weekend everywhere in the world (Thursday, Friday in the Middle East, Saturday, Sunday in the West, etc.).

I asked myself, why this was the case? After observing the characteristics noted above for a number of years, my opinion is that the efforts lacked an effective sales program. As a result, since the middle 1990's, I have been recommending that all site reliability efforts are integrated and definitely include an operations representative who can be a process engineer or senior operator or both. The reasons are because all site equipment reliability depends upon the process requirements and that, let's face it, the plant is run by operations and if there is an element of operations in the reliability group who agrees with the group's recommendation; acceptance and implementation have a much better chance. Because ... there is now a salesman in the reliability group!

Naturally the approach will be different in each plant and there are as many possible variations as there are plants. Figure 9.25 presents some of the possible structures of a site reliability initiative.

Site reliability initiative guidelines

- Specific, multi-discipline experienced reliability group
- A site culture change that makes reliability everyone's responsibility
- Operation, process and instrument reliability group members are on a one year rotation assignment

Figure 9.25 Site reliability initiative guidelines

Rotation of the operations, process and perhaps instrumentation members of the reliability group has shown to be a very good idea since returning members of the group to their own disciplines will naturally spread the word regarding function awareness and the importance of the program. It amounts to automatic function awareness training! Selection of the members of the group on rotation should be made carefully. I have found that these people should be the 'believable people' (experienced personnel who have the respect of their co-workers).

Remember again that regardless of make-up, the initiative must be multi-disciplined. The advantages of this arrangement are many. The major ones are presented in Figure 9.26.

Multi-disciplined reliability group advantages

- Valuable process information input
- Process and operation members are salesmen
- Significantly greater degree of function awareness of site equipment among all site disciplines
- Higher degree of ownership among all site disciplines
- Less finger pointing when problems occur

Figure 9.26 Multi-disciplined reliability group advantages

It is interesting to note that I have always observed that smaller remote plants (Artic region, New Zealand, Southern Chile and Platforms) have consistently out performed larger units in terms of action plan implementation. In these locations, they have to work together! And when they do, since everyone learns something regarding other discipline responsibilities and work scope, the results and acceptance of action plans flows smoothly.

Once the decision is taken to include operations and maintenance groups in the reliability program, consideration should be given to establishing 'ownership' in these groups in regards to the program. I have found that awareness training combined with simple 'tools' to inspire implementation of CCM and PDM help immensely. Figure 9.27 offers some suggested 'tools'.

Tools of the program

Suggested items to assist in CCM and PDM by operations and maintenance

- Laser temperature monitor
- Vibration pen
- Digital data logger to dump local information into DCS
- Strobe - for monitoring pump flinger ring operation and coupling condition
- A piece of $^{3}/_{4}$" pipe - for listening

Figure 9.27 Tools of the program

Once the participants are selected and work descriptions are defined and assignments made, a proven method of documenting all results for measurement of improvement and management support must be established. Guidelines for reports are presented in Figure 9.28.

Reliability report guidelines

- MTBF, MTTR and bad actor trends
- Cost savings from elimination of bad actors
- Cost to correct bad actors
- Net savings
- Benchmark recommendations
- Continuous improvement recommendations concerning reduction of PM and required turnaround work

Figure 9.28 Reliability report guidelines

Obtain and maintain management support

As noted in Figure 9.28, show net savings to the plant and believe me, you will obtain and maintain management support. However, you must show continuous savings and improvement. The reliability effort that only issues interesting newsletters won't be around too long. Management is interested in hearing the advice presented in Figure 9.29.

What plant management wants

- What was the problem?
- Is it permanently solved?
- How do you know it is solved? (use benchmark)
- How much did it cost to fix?
- What were our annual losses prior to the fix?
- How much did we save?

Figure 9.29 What plant management wants

Be sure to always provide proof (a benchmark) that the implemented solution has been applied elsewhere and has worked. Be prepared to back up the contention with written support and even a site visit to the benchmark facility. Do not forget to prepare a written agenda and forward it to the facility to be visited well in advance. Never expose yourself to loss of creditability with your management or another facility. You may work there someday!

As the program grows with success, use its cost saving results to justify sufficient resources of material, group state of the art training and manpower. Once success is realized, management will be quick to press for more savings. They must be made aware that the success to date was the result of the workload, manpower and experience level of the group. As work increases, the work to manpower and experience ratios must remain constant and new methods must be introduced to the group to increase effectiveness. The final section of this chapter presents guidelines for continuous improvement of this effort.

How to maintain continuous improvement of the established program

As I have previously stated, I have seen many an effective program fade with time. In order to sustain and improve an effective program, management support must be maintained, regardless of the business climate or management style. Some of the observed reasons for failure of reliability efforts are presented in Figure 9.30.

Why effective reliability programs do not continue

- Lack of management support due to:
  - Poor reports
  - Poor net saving results
  - Failure to benchmark recommendations
  - Lack of group experience
  - Change of business climate (poor corporate results)
  - Management change and not identifying with the new management team

Figure 9.30 Why effective reliability programs do not continue

Continued management support is earned and will yield results that will allow the reliability effort to prosper. Some of the advantages gained from a successful program, continuously supported by management, that will improve the program and maintain its reputation are shown in Figure 9.31.

Rewards for an effective reliability effort

- The percentage of proposed action plans approaches 100%
- Approval for procurement of additional 'reliability tools'
- Approval for site specific awareness training
- Approval for yearly vendor continuous improvement meetings
- Approval for attendance at industry seminars
- Approval for participation in user groups
- Approval for independent audits performed for difficult reliability issues

Figure 9.31 Rewards for an effective reliability effort

Details concerning each of the items in Figure 9.31 are presented below:

The percentage of proposed action plan implementation approaches 100%

This is the main objective of any reliability effort and it will yield many additional opportunities for personal professional growth, plant reputation for reliability and excellence and most importantly, program creditability. But it must be maintained and to be maintained, the group must truly grow in terms of tools, techniques and vendor communication.

Approval for procurement of additional 'reliability tools'

The effectiveness of all reliability efforts are based on the program's ability to obtain state of the art hardware and software. Given today's rapid rate of change, requests must be approved promptly by management. Be sure to keep track of the money saved by the program and use this figure when requesting additional reliability tools. Also do not forget to list the potential additional savings that the new equipment can produce.

Approval for site specific awareness training of all groups

Even if management support is obtained for site and off site training, it may not be approved if the requests are cycled through the human resources department. It must be appreciated that this department, like all other groups on site, have their budgets and culture. I have continuously witnessed proven valuable training workshops rejected by the HR department on the basis of cost and I have also observed that towards the end of the budget year that training is often selected for no apparent reason – other than the budget must be used or it won't be available next year. My recommendations for the improved acceptance of training programs are noted in Figure 9.32.

How to obtain training approval

- Justify on the basis of potential savings
- Benchmark people who have taken the program
- Recommend to your immediate supervisor
- Pass the name of the plant manager and your immediate supervisor to the organization that will conduct the training
- Work towards a joint effort between HR and the client (group that will be trained) in coordinating training

Figure 9.32 How to obtain training approval

You really cannot blame the human resources department if a proper objective for approving training is not given to them. Recommend to top plant management that training decisions be a joint effort of the concerned department (maintenance, technical, operations etc. and the human resources department).

Approval for yearly vendor continuous improvement meetings

It is truly amazing to me how, after the equipment is started, ties are severed with the equipment supplier. Perhaps, it is because of fear of high prices for spare parts or warranty rejection. However considering the exposure to revenue and profit losses that the end user faces, especially today with the size of process unit growing larger and larger, strong consideration should be given to increased vendor – end user communication after the order. The guidelines for vendor/end user communication guidelines are shown in Figure 9.33.

Vendor/end user communication guidelines

- Consider 'e based' vendor communication systems
- List problems, questions and suggestions and inform vendor
- Assure that all vendor updates are read and considered
- Hold a yearly vendor continuous improvement meeting, at the vendor's offices as a minimum
- Always provide detailed agendas for any vendor meetings well in advance.

Figure 9.33 Vendor/end user communication guidelines

Approval for attendance at industry seminars

Industry seminars are excellent opportunities for technology transfer if ... you are sent to the seminar and if management allows implementation of the tremendous amount of new knowledge you will bring back to the plant. However, if management has already bought in to your program, you can be sure that these principles will be implemented.

Approval for participation in user groups

User groups seem to be on the decline. Perhaps our friends in the legal profession have had something to do with that! It is true that many user groups have members that are potential competitors and may have issues. However, the Gas Turbine Users Association (GTUA) and the Texas A&M Turbomachinery Symposium are but two examples where competitors rub elbows and even present together! Inform management of the benefits of participation in user groups ... in terms of potential increased profit!

Approval for independent audits performed for difficult reliability issues

A final benefit of obtaining creditability of your reliability program is to be able to call in external help for a true ‘global based audit’ when required to solve long term, costly reliability issues. If you have not truly established the creditability of your program with management, this action will be impossible.

In summary, the objective of this chapter was to present practical guidelines for establishing, maintaining and continually improving a world class site reliability effort. Remember that success is dynamic. Change and modifications to the program are always necessary but they will require continued management support no matter who the manager is!

Appendix:

Questionnaires

Rotating equipment continuous improvement study engineering/ maintenance input data questionnaire

I. General

1. Please provide organization charts for maintenance and engineering groups.
2. Please provide start-up date for each process unit and approximate dates of T & I's (turnarounds) since start-up.
3. Please provide approximate T & I's (turnaround) downtime periods (projected and actual) for last two (2) T & I's.
4. List, if possible, reasons for extension of projected T & I's (turnaround) downtime period.
5. Do maintenance personnel become involved with determination of root cause problems? Yes ____ No ____
6. Since start-up of each process unit, have there been changes concerning the maintenance performed on-site? Yes ____ No ____ (Please list for each area)
7. Please describe specific maintenance practices for rotating equipment performed:

On-site	*Off-site*

   *Please note location

8. Please list approximate cost of maintenance labor.
9. Please describe alignment method for:
   - Critical equipment
   - General purpose equipment.
10. Please describe any centrifugal compressor fouling problems and methods used to prevent fouling and to clean compressors.
11. Please describe transient vibration data obtained during start-up of critical equipment (bode, cascade, polar).

12. Please describe if the following at speed data is obtained for critical unspared equipment:
    A. Unfiltered vibration
    B. Radial bearing vibration orbits
    C. Radial shaft position
    D. Bearing pad temperatures
    E. Axial position
    F. Thrust pad temperatures
    G. Centrifugal compressor balance line ΔP
13. Please describe what items are checked during bearing replacement.
14. Please describe what items are checked during pump mechanical seal replacement.
15. Have there been any instances of critical equipment coupling failure? If so, please describe specifics (use separate page).
16. Please briefly describe problems with mechanical/hydraulic governor systems (TG-10, 13, 17) for general purpose turbines.

17 Please briefly describe problems with critical steam turbine governor systems (PGPL or electro-hydraulic).

18. How often are turbine overspeed trips checked?
19. Please describe any problems with critical steam turbine trip solenoids.
20. Please describe any problems with oil contamination (water) for single stage (general purpose) steam turbines.
21. Please describe types of mechanical seal quench systems and seal jacket cooling systems used for bottoms, HGO and LGO pumps (provide P & ID if possible).
22. Please provide reports for bad actor reliability problems. (Major compressor trains, caustic pump mechanical seal problems, bad actor mechanical seals)
23. Please list what information you could obtain regarding process and operation that would enable you to perform your job better.
24. Please provide example of a typical failure analysis report.
25. Please provide present PM (preventive maintenance) schedule for:
    - Single stage process pump
    - Motor (greased anti friction bearings)
    - Motor (sleeve bearings)
    - Centrifugal compressor train
    - Reciprocating compressor unit

II. Specific questions for each area

Note: Please copy this form and distribute to the appropriate personnel in each designated area of the audit.

1. Condition monitoring data

A. Method (i.e. manual vs. 'data trap' vs. D.C.S.)

Example	Method	Years used
	Manual logs	

B. Condition monitoring parameters. (Note) Please answer if the following parameters are condition monitored (Note 1)

1. Centrifugal compressor performance parameters (P_1, T_1, P_2, T_2 flow, speed, gas analysis)		Yes	No
2. Pump performance parameters (P_1, P_2, S.G., flow speed)		Yes	No
3. Compresor seal fluid (gas or oil) supply flows (or valve position if oil system)	(Gas)	Yes	No
	(Oil)	Yes	No
4. Centrifugal pump seal stuffing box pressures and temperatures		Yes	No
5. Centrifugal compressor balance line ΔP		Yes	No
6. Reciprocating compressor performance		Yes	No
7. Reciprocating compressor crankcase vibration and rod drop		Yes	No
8. Reciprocating compressor jacket water temperature		Yes	No
9. Reciprocating compressor packing leakage		Yes	No
10. Reciprocating compressor lubrication (packing, cylinder) usage		Yes	No
11. Centrifugal compressor seal oil trap leakage		Yes	No
12. Centrifugal compressor seal oil reservoir flash point		Yes	No
13. Steam turbine performance		Yes	No
14. Steam turbine seal leakage		Yes	No
15. Steam turbine T & T valve exercising		Yes	No
16. Prove on line the ability of the auxiliary oil pumps to start without unit trip			

Note 1. All questions in item B pertain only to equipment in each process unit

of the audit. It is suggested that copies of this form be distributed to personnel in each area concerned with the audit.

Note 2. In the space provided between items state item numbers of units where the answer is 'yes'.

Note 3. It is assumed all centrifugal compressor units are vibration and bearing temperature condition monitored. Please advise if this is not the case.

2. Preventive maintenance data

Refer to Note 1 in item 1 above.

A. Please answer if the following items are changed on a preventive (time interval) basis. If the answer is yes, please state the time between replacement (Δ Time)

		ΔTime		
1.	Compressor rotors	____	Yes	No
2.	Compressor bearing and turbine	____	Yes	No
3.	Compressor seals	____	Yes	No
4.	Compressor labyrinths	____	Yes	No
5.	Steam turbine rotor (spared turbines)	____	Yes	No
6.	Steam turbine rotor (unspared turbines)	____	Yes	No
7.	Pump bearings	____	Yes	No
8.	Pump seals	____	Yes	No
9.	Pump wear rings	____	Yes	No
10.	Electrical switches	____	Yes	No
11.	Accumulator bladders	____	Yes	No
12.	Control valve diaphragms	____	Yes	No
13.	Oil filter cartridges	____	Yes	No

B. Please answer if the following items are cleaned on a preventive basis.

1.	Oil reservoirs	____	Yes	No
2.	Cooler tubes	____	Yes	No
3.	Control valve pulsation dampeners (snubbers)	____	Yes	No
4.	Seal oil drain types	____	Yes	No
5.	Seal oil degassing tanks	____	Yes	No
6.	Reciprocating compressor lubricators	____	Yes	No

3. Turnaround activities

Please list turnaround activities for each critical (unspared) rotating equipment unit (this includes driver, driven, transmission devices and auxiliary systems) on the following form

Area **Train** ____________

Type and items no.	T & I dates	Major item replaced (note 1)	Reason (note 2)	Part cost	Labor time (hours)	Did activity extend T & I (note 3)

Note 1. Major items: rotors, internals, coupling, bearings, seals, control valves

Note 2. Use following code

P.M. = preventive maintenance

P.D.M. = predictive maintenance indicated imminent failure

F.P. = failed part

Note 3. If activity extended T & I, indicate how many days.

4. Critical equipment forced outages

Please complete the following form for forced outages of major unspared equipment.

Equip. type & item no.	Date of event	Reason for outage	Parts affected	Process unit outage hours	Time to repair (hours)	Cost of parts	Suspected causes	Corrective action

5. Please list the field monitoring performed by operations and note if it is MRC (control room) and/or local.

Type	Parameters measured	Control room	Local
Pumps			
Steam turbines			
Compressors			
Lube/seal systems			

Rotating equipment continuous improvement study operations engineering and operations input data questionnaire

I. General

1. Please provide technical organization chart for:
   - Operations
   - Operations engineering
2. Please provide approximate daily revenue loss (local currency) for all the major process units in the plant.
3. Please provide information concerning control room modernization (DCS) type, date, etc.
4. Please list the field monitoring performed by operations and note if it is MRC (control room) and/or local.

Type	Parameters measured	Control room	Local
Pumps			
Steam turbines			
Compressors			
Lube/seal systems			

5. Do operations engineers and operations personnel become involved with root cause determination of rotating equipment problems?
6. Please describe any centrifugal compressor fouling problems.
   - Methods used to prevent
   - Cleaning methods

7. Please list what information you could obtain regarding rotating equipment, that would enable you to perform your job better.
8. Please advise if and when all site major centrifugal compressors (list each compressor) are checked for performance (head and efficiency).

II. Specific questions for each area

Note: Please copy this form and distribute it to the appropriate personnel in each designated area of the audit

1. Rotating equipment process and operation questions

   Please complete the following questions for each critical (unspared) equipment item in your area. Make additional copies of this form as required.

   1. Centrifugal compressor process conditions
      - A. Does compressor surge on start-up?
        - Molecular weight, gas temperature charges
      - B. Does compressor surge during normal operations?
      - C. Any problems with moisture in the process?
      - D. Has performance (flow) been decreasing with time or has turbine speed been increasing with time?
      - E. Compressor suction strainer problems?
   2. Steam conditions for critical steam turbines
      - A. Present normal steam conditions and flow rates.
      - B. Does 'off design operation occur' (low inlet pressure or temperature or high exhaust pressure?
      - C. Any slugs of moisture go through the turbine?
      - D. Any fouling of turbine?
      - E. Trend of first stage pressure vs time.
      - F. Any steam strainer problems?
   3. Alarms or trips
      - A. Vibration
      - B. Bearing temperature
      - C. Thrust bearing position
      - D. Overspeed
      - E. High separator liquid level

    F. Low seal oil Press
    G. Auxiliary pump running
    H. High oil temperature
    I. High oil filter differential

4. Seal oil drainers and degassing tank
    A. Do drainers ever need to be removed from service or bypassed?
    B. Any change in drainer flow with change in operation conditions?
    C. Any problems with degassing tank?
5. Steam turbine operation (critical/unspared)
    A. Any vibration problems during start-up?
    B. Any problems with steam seal system?
    C. Any problems with main condenser system?
    D. Any governor control problems?
    E. Can you manually exercise trip and throttle valve? If so, how often?
    F. Any problems with turning gear?
    G. Any problems with turbine tripping off the line or not tripping when signal is given?
6. Operation problems
    A. Is seal oil system for compressors put into operation before process gas is put into the case?
    B. Please describe any compressor operation problems.
7. Lube, control and seal oil problems
    A. Oil reservoir
        1. Oil sample results
        2. Low level alarms
    B. Pumps
        1. Cavitation noise
        2. Vibration
        3. Bearing or seal problems
        4. Turbine control problems
            - Speed decrease
            - Governor instability (hunting)

C. Trip or transfer of lube or seal oil pumps
    1. Can main pump be tripped and auxiliary pump start without tripping unit?
    2. Is this ever attempted during operations?
    3. What checks are performed during operation that auxiliary pump will start without tripping unit?

D. Relief valves
    1. Popping
    2. Not closing
    3. Improper setting

E. Transfer valves
    1. Any plug or handle problems?

F. Coolers
    1. Any problems with maintaining required oil supply temperature?

G. Filters
    1. Have filter cartridges ever been damaged during filter changeover?
    2. How often are filters changed?

H. Valve and controllers
    1. Any valve or controller instability?
        - On start-up
        - During normal operation
        - During pump changeover
        - When both lube or seal pumps are operating

8. Pump problems – 'process bad actors'

A. Please indicate pumps experiencing repeated seal replacements (more than 1 per year)

B. Please indicate pumps experiencing repeated bearing replacements (more than 1 per year)

C. Please indicate pumps that require internal inspection and/or replacement bearings, rings, internals, etc.

D. Please list any other pump problems.

9. Critical equipment forced outages

   Please complete the following form for forced outages of major unspared equipment.

OEU/Operations critical equipment forced outages

Equip. type and item no.	Date of event	Reason for outage	Parts affected	Process unit outage hours	Time to repair (hours)	Cost of parts	Suspected causes	Corrective action

Design Audit Example 1

Rotor axial (thrust) forces tutorial

In every rotating machine utilizing reaction type blading, a significant thrust is developed across the rotor by the action of the impellers or blades. Also in the case of equipment incorporating higher than atmospheric suction pressure, a thrust force is exerted in the axial direction as a result of the pressure differential between the pressure in the case and atmospheric pressure.

In this section we will cover a specific rotor thrust example and calculate thrust balance for a specific case. We will see the necessity in some applications of employing an axial force balance device known as a balance drum. In many instances, the absence of this device will result in excessive axial (thrust) bearing loadings. For the case of a machine with a balance device, the maintenance of the clearances on this device are of utmost importance. In many older designs the clearances are maintained by a fixed close clearance bushing made out of Babbitt which has a melting temperature of approximately 350°F, depending on the pressure differential across the balance drum. If the temperature in this region should exceed this value, the effectiveness of the balance drum would suddenly be lost and catastrophic failures can occur inside the machine. Understanding the function of this device and the potential high axial forces involved in its absence is a very important aspect of condition monitoring of turbo-compressors.

We will also examine various machine configurations including natural balanced (opposed) thrust and see how thrust values change even in the case of a balanced machine as a function of machine flow rate.

Finally, we will examine thrust system condition monitoring and discuss some of the confusion that results with monitoring these machines.

The hydrodynamic thrust bearing

A typical hydrodynamic double acting thrust bearing is pictured in Figure 9.34.

The thrust bearing assembly consists of a thrust collar mounted on the rotor and two sets of thrust pads (usually identical in capacity) supported by a base ring (Michell type).

The Kingsbury type includes a set of leveling plates between each set of pads and the base ring. This design is shown in Figure 9.35.

Both the Michell and Kingsbury types are used. Figure 9.36 provides a

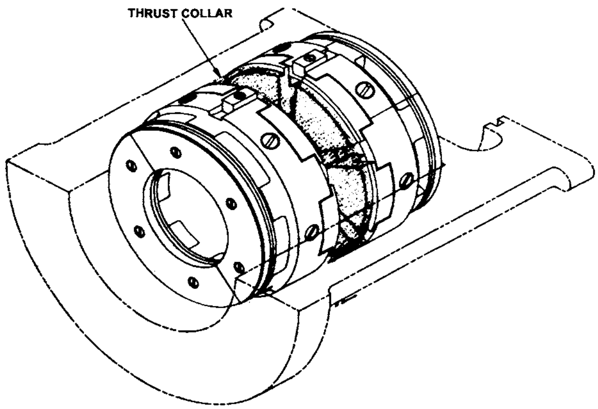

THRUST COLLAR

Figure 9.34 Double acting self equalizing thrust bearing assembly (thrust collar removed) (Courtesy of Elliott Company)

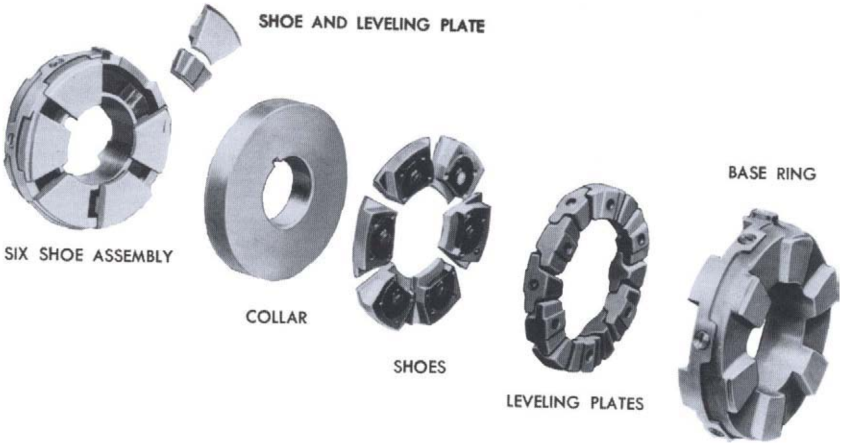

SHOE AND LEVELING PLATE
SIX SHOE ASSEMBLY
COLLAR
SHOES
LEVELING PLATES
BASE RING

Figure 9.35 Small Kingsbury six-shoe two-direction thrust bearing. Left-hand group assembled, except for one shoe and 'upper' leveling plate. Right-hand group disassembled. (Courtesy of Kingsbury, Inc)

view of the leveling plates providing the self-equalizing feature in the Kingsbury design. The self-equalizing feature allows the thrust pads to lie in a plane parallel to the thrust collar.

Regardless of the design features, the functions of all thrust bearings are:

- To continuously support all axial loads
- To maintain the axial position of the rotor

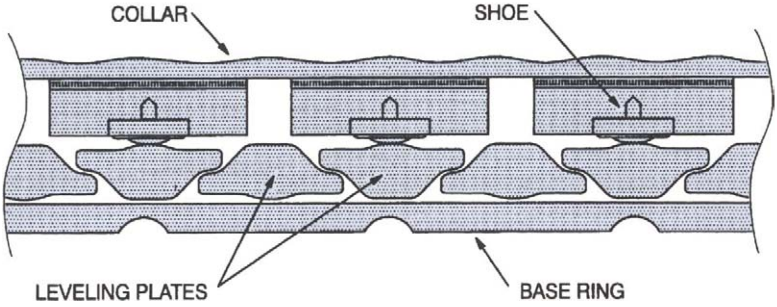


Figure 9.36 Self-equalizing tilt-pad thrust bearing (View – looking down on assembly) (Courtesy of Kingsbury, Inc)

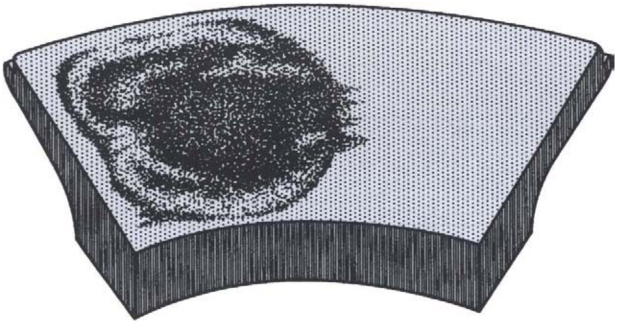


Figure 9.37 Evidence of overload on a tilt-pad self-equalizing thrust bearing pad (Courtesy of Kingsbury Corp)

The first function is accomplished by designing the thrust bearing to provide sufficient thrust area to absorb all thrust loads without exceeding the support film (oil) pressure limit (approximately 500 psi).

Figure 9.37 shows what occurs when the support film pressure limit is exceeded.

The oil film breaks down, thus allowing contact between the steel thrust collar and soft thrust bearing pad overlay (Babbitt). Once this thin layer (1/16") is worn away, steel to steel contact occurs resulting in significant turbo-compressor damage.

Thrust pad temperature sensors, located directly behind the Babbitt at the pad maximum load point protect the compressor by tripping the unit before steel to steel contact can occur.

Figure 9.38 presents different Kingsbury bearing size rated capacities as a function of speed.

Figure 9.39 shows how thrust pad temperature and thrust load are related for a given thrust bearing size and shaft speed. Note that the

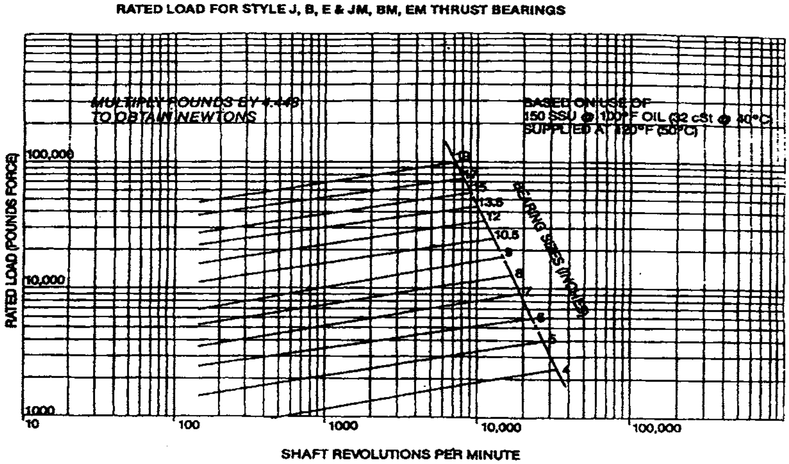


Figure 9.38 Thrust bearing rated load vs. speed (Courtesy of Kingsbury, Inc)

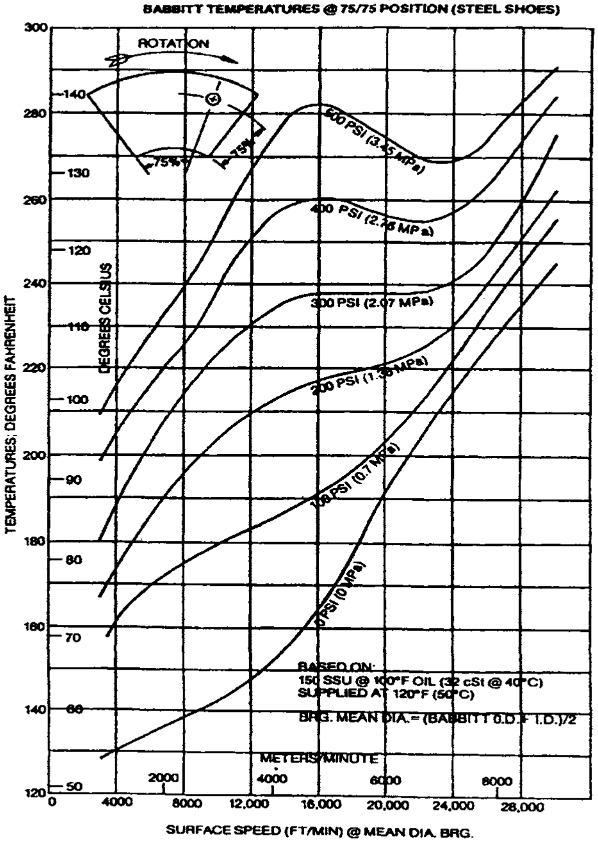


Figure 9.39 Relationship between thrust pad temperature and thrust load (Courtesy of Kingsbury, Inc.)

greater the thrust load (P.S.I.), the smaller the oil film and the greater the effect of oil viscosity on oil flow and heat removal. Based on a maximum load of 500 psi, it can be seen from Figure 9.39 that a turbo-compressor thrust bearing pad temperature trip setting should be between 260° and 270°F.

Other than to support the rotor in an axial direction, the other function of the thrust bearing is to continuously maintain the axial position of the rotor. This is accomplished by locating stainless steel shims between the thrust bearing assembly and compressor axial bearing support plates. The most common thrust assembly clearance with the thrust shims installed is 0.011 – 0.014. These values vary with thrust bearing size. The vendor instruction book must be consulted to determine the proper clearance.

The following procedure is used to assure that the rotor is properly positioned in the axial direction.

1. With thrust shims removed, record total end float by pushing rotor axially in both directions (typically .250" – .500").
2. Position rotor as stated in instruction book.
3. Install minimum number of stainless steel thrust shims to limit end float to specified value.*

*An excessive number of thrust shims act as a spring resulting in a greater than specified axial clearance during full thrust load conditions.

Proper running position of the rotor is critical to obtaining optimum efficiency and presenting axial rubs during transient and upset conditions (start-up, surge, etc.)

Impeller thrust forces

Every reaction type compressor blade set or impeller produces an axial force towards the suction of the blade or impeller. Refer to Figure 9.40.

Note that the pressure behind the impeller is essentially constant (50 psi), but the pressure on the front side of impeller varies (from 50 psi to 40 psi) because of the pressure drop across the eye labyrinth. Every impeller in a multistage compressor will produce a specific value of axial force towards its suction at a specific flow rate, speed and gas composition. A change in any or all of these parameters will produce a corresponding change in impeller thrust.

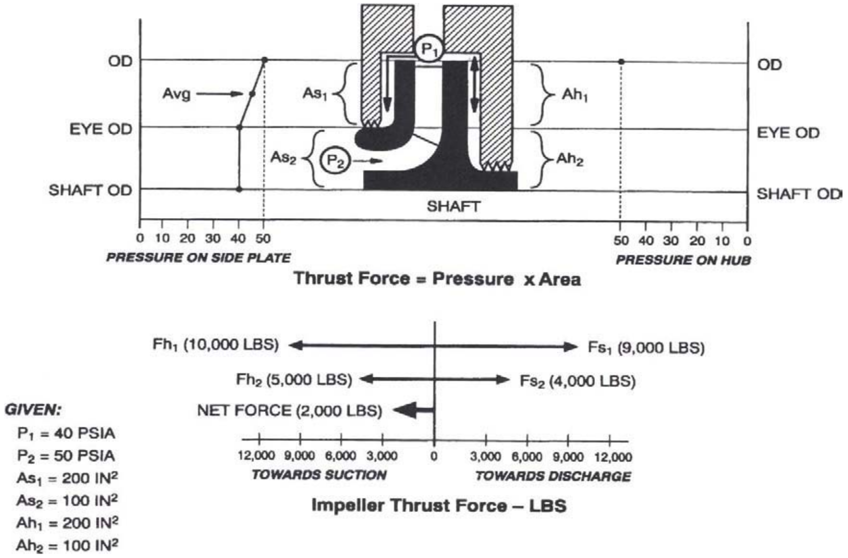

OD
Avg
EYE OD
SHAFT OD
As₁
As₂
P₁
P₂
Ah₁
Ah₂
SHAFT
OD
EYE OD
SHAFT OD
0 10 20 30 40 50
PRESSURE ON SIDE PLATE
50 40 30 20 10 0
PRESSURE ON HUB
Thrust Force = Pressure x Area
Fh₁ (10,000 LBS)
Fs₁ (9,000 LBS)
Fh₂ (5,000 LBS)
Fs₂ (4,000 LBS)
NET FORCE (2,000 LBS)
12,000 9,000 6,000 3,000 0 3,000 6,000 9,000 12,000
TOWARDS SUCTION
TOWARDS DISCHARGE
Impeller Thrust Force – LBS
GIVEN:
P₁ = 40 PSIA
P₂ = 50 PSIA
As₁ = 200 IN²
As₂ = 100 IN²
Ah₁ = 200 IN²
Ah₂ = 100 IN²

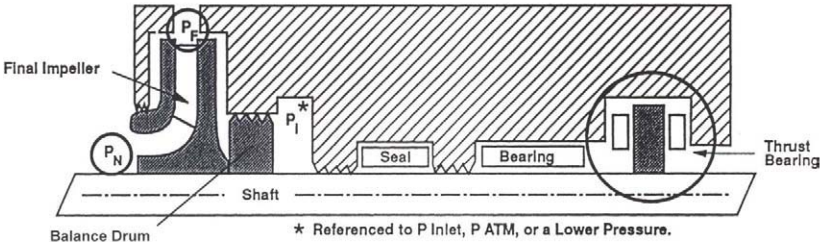
Figure 9.40 Impeller thrust force

Rotor thrust balance

Figure 9.41 shows how a balance drum or opposed impeller design reduces thrust force.

The total impeller force is the sum of the forces from the individual impellers. If the suction side of the impellers is opposed, as noted in Figure 9.41, the thrust force will be significantly reduced and can approach 0. If the suction side of all impellers are the same (in series), the total impeller thrust force can be very high and may exceed the thrust bearing rating. If this is the case, a balance drum must be mounted on the rotor as shown in Figure 9.41. The balance drum face area is varied such that the opposing force generated by the balance drum reduces the thrust bearing load to an acceptable value. The opposing thrust force results from the differential between compressor discharge pressure (P_F) and compressor suction pressure (P_1) since the area behind the balance drum is usually referenced to the suction of the compressor. This is accomplished by a pipe that connects this chamber to the compressor suction. This line is typically called the 'balance line'.

It is very important to note that a balance drum is used only where the thrust bearing does not have sufficient capacity to absorb the total compressor axial load. And the effectiveness of the balance drum depends directly on the balance drum seal. Fail the seal, (open clearance significantly) and thrust bearing failure can result.


Final Impeller
P_F
P_I*
P_N
Seal
Bearing
Thrust Bearing
Shaft
Balance Drum
* Referenced to P Inlet, P ATM, or a Lower Pressure.

Total Impeller Thrust (LB) = Σ Individual Impeller Thrust
Balance Drum Thrust (LB) = (P_F - P_I) x (Balance Drum Area)
Thrust Bearing Load (LB) = Total Impeller Thrust – Balance Drum Thrust

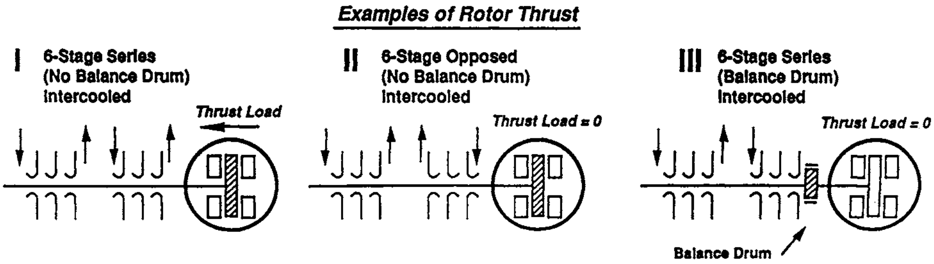

Examples of Rotor Thrust
I 6-Stage Series (No Balance Drum) Intercooled
Thrust Load
II 6-Stage Opposed (No Balance Drum) Intercooled
Thrust Load = 0
III 6-Stage Series (Balance Drum) Intercooled
Thrust Load = 0
Balance Drum

Figure 9.41 Rotor thrust force

A common misunderstanding associated with balance drum systems is that a balance drum always reduces the rotor thrust to zero. Refer to Figure 9.42 and observe that this statement may or may not be true depending on the thrust balance system design. And even if it is, the thrust is zero only at one set of operating conditions.

Figure 9.42 shows a rotor system designed four (4) different ways. Note how the thrust **always** changes with the flow rate regardless of the design. Another misconception regarding thrust balance systems is the normal or ‘active’ direction of thrust. In many cases, the active thrust is assumed to always be towards the suction of the compressor. Observing Figure 9.42, it is obvious that the ‘active’ direction can change when the turbo-compressor has a balance drum or is an opposed design. It is recommended that the use of active thrust be avoided where possible and that axial displacement monitors be labeled to allow determination of the thrust direction at all times.

Please refer to Figure 9.43 which shows a typical thrust displacement monitor.

These monitors detect thrust position by targeting the shaft end, thrust collar or other collar on the rotor. Usually two or three probes (multiple voting arrangement) are provided to eliminate unnecessary compressor trips. The output of the probes is noted on the monitor as

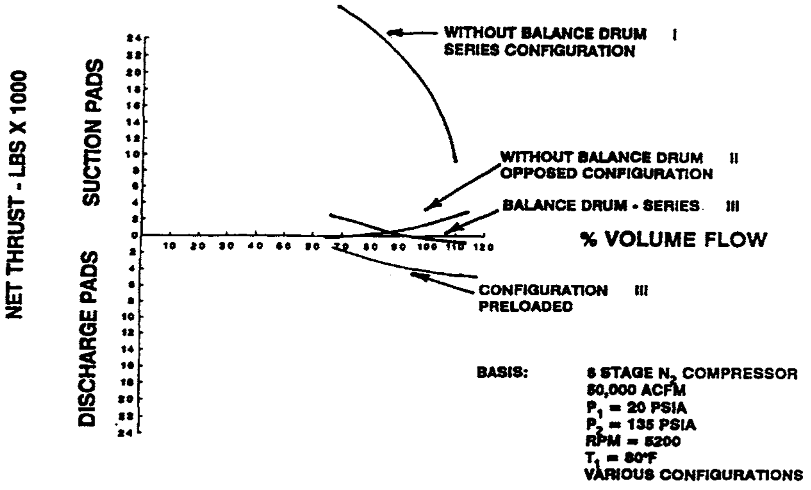


Figure 9.42 Net thrust load vs. % flow

either + (normal) or – (counter). However, this information gives no direct indication of the axial direction of the thrust collar. The following procedure is recommended:

1. With compressor shutdown, push rotor towards the suction and note direction of displacement indicator.
2. Label indicator to show direction towards suction of compressor.

Knowing the actual direction of the thrust can be very useful during troubleshooting exercises in determining the root cause of thrust position changes.

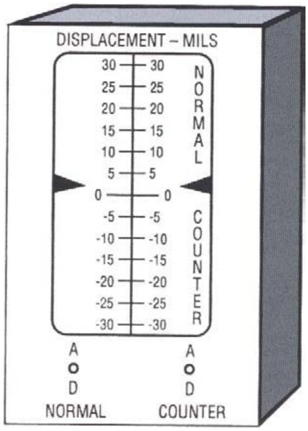


Figure 9.43 Axial thrust monitor

Thrust condition monitoring

Failure of a thrust bearing can cause long term and possibly catastrophic damage to a turbo-compressor. Condition monitoring and trending of critical thrust bearing parameters will optimize turbo-compressor reliability.

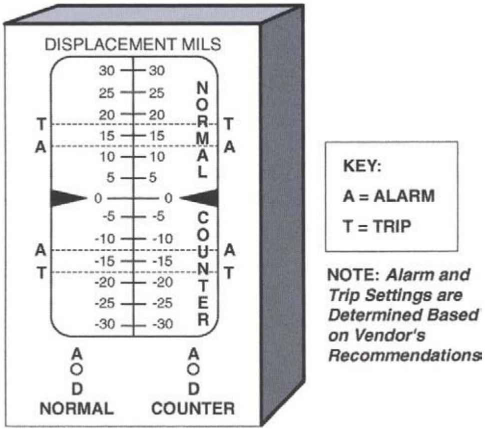

DISPLACEMENT MILS
30
25
20
15
10
5
0
-5
-10
-15
-20
-25
-30
NORMAL
COUNTER
T
A
A
T
AOD NORMAL
AOD COUNTER
KEY:
A = ALARM
T = TRIP
NOTE: Alarm and Trip Settings are Determined Based on Vendor's Recommendations

Figure 9.44 Axial displacement condition monitoring

The critical thrust bearing condition monitoring parameters are:

- Rotor position
- Thrust pad temperature
- Balance line ΔP

Rotor position is the most common thrust bearing condition parameter and provides useful information regarding the direction of thrust. It also provides an indication of thrust load but does not confirm that thrust load is high. Refer to Figure 9.44.

All axial displacement monitors have pre-set (adjustable) values for alarm and trip in both thrust directions. Typically, the established procedure is to record the thrust clearance (shims installed) during shutdown and set the alarm and trip settings as follows:

$$\text{Alarm} = \frac{\text{clearance}}{2} + 10 \text{ mils (each direction)}$$

$$\text{Trip} = \text{alarm setting} + 5 \text{ mils (each direction)}$$

The above procedure assumes the rotor is in the mid or zero position of the thrust clearance. An alternative method is to hand push the rotor to the assumed active position and add appropriate values for alarm and trip.

The writer personally recommends the first method since an active direction of thrust does not have to be assumed.

As noted, axial displacement monitors only indicate the quantity of thrust load. False indication of alarm or even trip settings can come from:

- Compression of thrust bearing components
- Thermal expansion of probe adaptors or bearing brackets
- Loose probes

It is strongly recommended that any alarm or trip displacement value be confirmed by thrust pad temperature if possible prior to taking action. Please refer back to Figure 9.39 and note that the thrust pad temperature in the case of thrust pad overload is approximately 250°F. If an axial displacement alarm or trip signal is activated observe the corresponding thrust pad temperature. If it is below 220°F, take the following action:

- Observe thrust pads. If no evidence of high load is observed (pad and back of pad) confirm calibration of thrust monitor and change settings if necessary.

The last condition monitoring parameter for the thrust system is balance line pressure drop. An increase of balance line ΔP will indicate increased balance drum seal leakage and will result in higher thrust bearing load. Noting the baseline ΔP of the balance line and trending this parameter will provide valuable information as to the root cause of a thrust bearing failure. In many field case histories, the end user made many thrust bearing replacements until an excessive balance drum clearance was discovered as the root cause of the thrust bearing failure. It is a good practice to always check the balance line ΔP after reported machine surge. Surging will cause high internal gas temperatures which can damage the balance drum seal.

This completes the section on thrust forces. An example of a balance drum calculation is given below.

Balance drum audit exercise

Title:	Determining the thrust load of a three (3) stage compressor and the effect of different balance drum diameters.
Required:	I. Calculate rotor thrust for example assuming no balance drum. II. Calculate rotor thrust assuming balance drum is installed on rotor.

Task 1 Given:

- Rotor and compressor internals shown in Figure 9.45

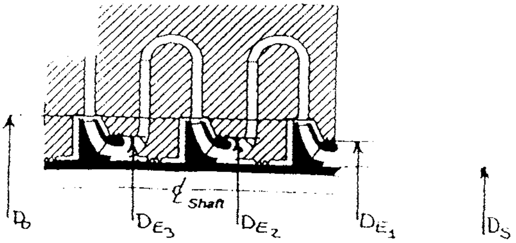


Figure 9.45

- D_0 of each impeller = 40in. diameter
- D_E of each impeller as follows:
  D_{E1} = 20 in. diameter
  D_{E2} = 18 in. diameter
  D_{E3} = 16 in. diameter
- D_S of shaft = 8 in. diameter

Task 1 Find: Rotor thrust for the following data.

Stage	1		2		3	
	P_1	P_2	P_1	P_2	P_1	P_2
56,000 acfm	20	27	27	38	38	50
51,000 acfm	20	30	30	43	43	57
42,000 acfm	20	33	33	47	47	62

Note: all pressures in psia

Task 1 Procedure:

1 Relationship:
Individual impeller thrust:

$$F_{T_N} = [P_{2_N} - P_{1_N}] \bullet [(\pi/4) \bullet [(D^2_{E_N} - D^2_S)]$$

Where: F = Impeller thrust force (lb)
N = Impeller stage
D_E = Impeller eye diameter (inches)
D_S= shaft diameter inches

Stage 1 $F_{T_1} = [P_{2_1} - P_{1_1}] \bullet \pi/4 \bullet [(D^2_{E_1} - D^2_S)]$

$2639_1 = [(30) - (20)] \bullet \pi/4 \bullet [(20)^2 - (8)^2]$

Stage 2 $2654_2 = [(43) - (30)] \bullet \pi/4 \bullet [(18)^2 - (8)^2]$

Stage 3 $2111_3 = [(57) - (43)] \bullet \pi/4 \bullet [(16)^2 - (8)^2]$

Note: obtain values of P_{2_N}, P_{1_N} from stage calculations.

2 Relationship:
Total impeller thrust

$$F_{TOTAL} = \sum_1^N F_{T_N}$$

$$= F_{T_1} + F_{T_2} + F_{T_3}$$

Where + = thrust towards impeller suction

Where – = thrust towards impeller discharge

$7404\ \text{LB} = 2639_1 + 2654_2 + 2111_3$

Note: This relationship assumes that impeller discharge pressure is equal on the hub (backside) and eye (front side) of each impeller. Due to disc friction (windage) and impeller seal leakage this is not exactly the case. Therefore, the actual impeller thrust towards the suction of each impeller is slightly greater.

Task 2 Find: Rotor thrust for 51,000 acfm flow and P_{2_N}, P_{1_N} values in Task 1 assuming an 18" balance drum is installed on rotor.

Plot results on curve on Figure 9.47.

Task 2 Given:

- Rotor and compressor internals shown below.

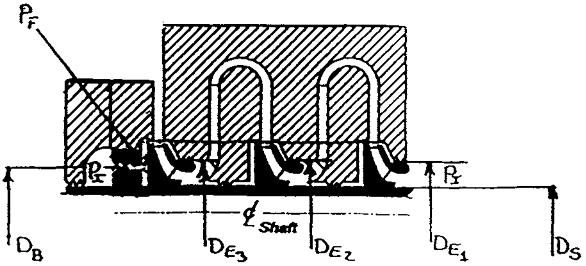


Figure 9.46

- D_0 of each impeller = 40in. diameter

  Where: D_B = Balance drum = 18"

Task 2 Procedure:

1 Relationship:

$$F_{BAL\ DRUM} = (P_F - P_S) \bullet (A_{BAL\ DRUM})$$

$$7555\ \text{LB} = [(57) - (20)] \bullet [\pi/4\ ((18)^2 - (8)^2)]$$

Where: P_F = Final compressor discharge pressure
P_S = Compressor suction pressure
$A_{BAL\ DRUM} = \pi/4\ (D^2_{BD} - D^2_S)$

2 Relationship:

$F_{\text{THRUST BEARING}} = F_{\text{IMPELLERS}} - F_{\text{BALANCE DRUM}}$

151 (lb) = 7404 (lb) -7555 (lb)

Plot values on curve on Figure 9.47.

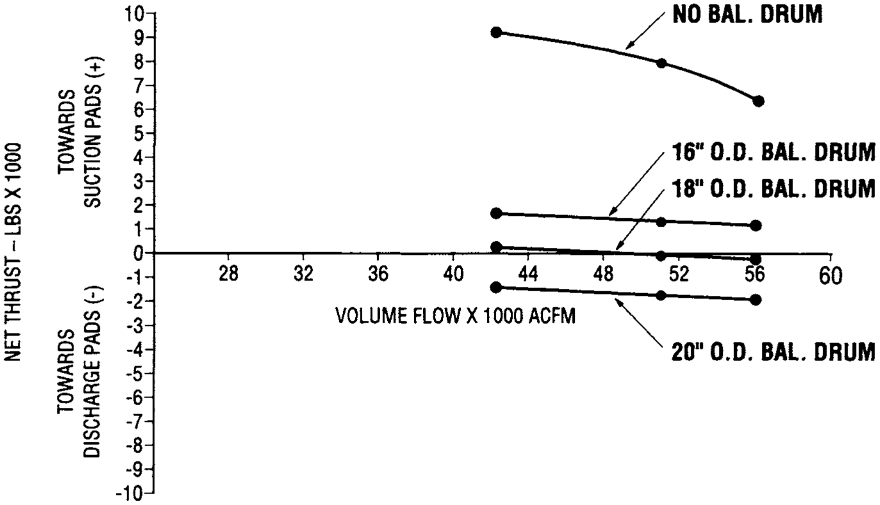


Figure 9.47 Compressor Thrust vs. inlet flow for no balance drum and various balance drum diameters

Design Audit Example 2

Lube system design tutorial

The lube system design audit objective is to confirm the scope of supply, component design and correct manufacture of the specified system. Two factors that play a great part in attaining a reliable auxiliary system are communication and revenue. Frequently auxiliary systems are subcontracted by the original equipment manufacturer to a speciality auxiliary system facility. The communication between original equipment manufacturer (OEM) and sub-vendor is not always efficient. After receipt of an order to manufacture an auxiliary system, confirmation of scope of supply should be obtained with the end manufacturer of the system present.

The other factor involved concerns the cost of the critical equipment and the price charged for that equipment. In the real world of competition, original equipment vendors many times sell the equipment for less than originally anticipated. Therefore, costs are high and profit decreases. In an effort to minimize cost, sub-systems can suffer in terms of design and quality. OEMs will competitively select sub-vendors for auxiliary systems. Care must be taken to assure the selected sub-vendor is an experienced, quality shop.

This section then will deal with the major areas of system design, the component sizing audit, system manufacturing and factory testing.

Design audit agenda

In this section, we will be dealing with the specific areas important to the confirmation of auxiliary system design and manufacture. To assure maximum effectiveness of these reviews, it is recommended that a prior agenda, mutually agreed upon between OEM and user, be generated and supplied to both parties well in advance of any meetings. In addition to detailing subjects of the discussion, the agenda should also define the attendees of the meeting. A well defined meeting is still ineffective if the participants are not familiar with the subject or have a minimum amount of experience.

Confirmation of scope

For the reasons mentioned above, scope review approximately one to two months after auxiliary system order placement is recommended. The major areas of scope review are:

A. Schematic review
B. Data sheet review
C. Exceptions to specification

Schematic (P & ID) review

The original system schematic (P & ID) (console and unit) as contained in the equipment specification should be reviewed at this point to confirm all system logic and instrumentation is as specified. That is, the schematic should be reviewed in the framework of P&ID (process and instrument diagram). All comments should be noted and the system schematics corrected.

Data sheet review

The system data sheet should be thoroughly reviewed and be complete at this point to include specific component details and desired manufacturers of major components. This review goes both ways. That is, vendor required information and user information must be detailed and correct on the data sheets. Frequently utility information and site information is not complete. This absence can only lead to reliability and communication problems in the field. As with any meeting, detailed minutes should be kept and every effort be expended to resolve all open items prior to conclusion of meeting. Postponing decisions only creates inefficiencies.

Exceptions to specifications

All vendor exceptions to specifications must be reviewed and either be accepted or rejected. The final, mutually agreed to list of vendor exceptions should become part of the job specification.

Component sizing audit

A typical component sizing audit form is included in the back of this section. We will now review the major areas of this audit form and comment relative to the specific items.

System requirements

The first subject of discussion concerns confirmation of auxiliary system flow rates, pressures and heat loads required. This information determines the size of all major system components. It must be correct and not subject to modification during the design process of the equipment. The component sizing agenda should emphasize the need

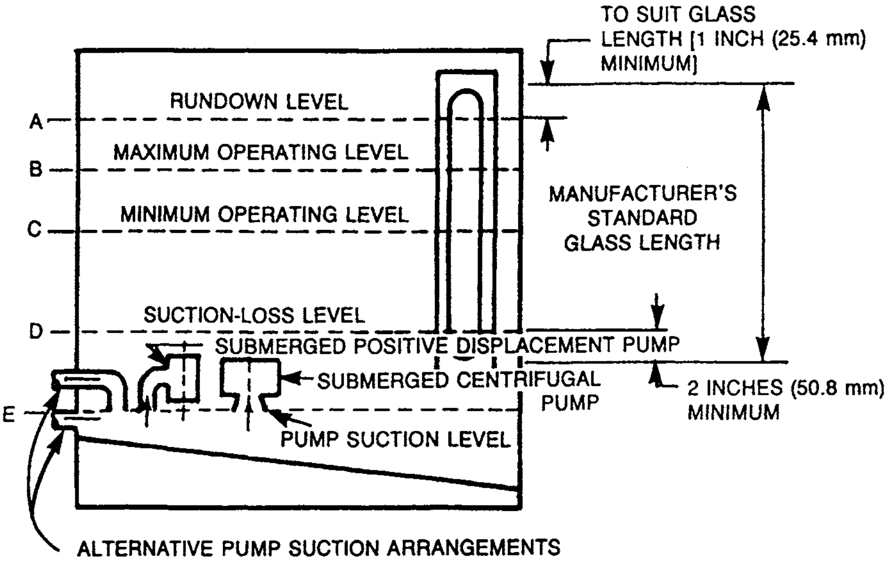

TO SUIT GLASS LENGTH [1 INCH (25.4 mm) MINIMUM]
A
RUNDOWN LEVEL
B
MAXIMUM OPERATING LEVEL
C
MINIMUM OPERATING LEVEL
MANUFACTURER'S STANDARD GLASS LENGTH
D
SUCTION-LOSS LEVEL
SUBMERGED POSITIVE DISPLACEMENT PUMP
SUBMERGED CENTRIFUGAL PUMP
2 INCHES (50.8 mm) MINIMUM
E
PUMP SUCTION LEVEL
ALTERNATIVE PUMP SUCTION ARRANGEMENTS

Figure 9.48 An auxiliary system reservoir

to have all required information furnished and confirmed by each critical equipment vendor. Attention is drawn to comparing values noted. If significant discrepancies appear, question them! Remember all critical equipment components are equivalent orifices and at a specified pressure will only pass a given flow. If the component oil flow specified is greater than the amount the components will actually pass, the excess oil will be bypassed back to the oil reservoir and could create overheating problems in the system. Conversely, if too low a value of component oil flow is specified, a system may continuously operate with both main and stand-by pump in operation since the capacity of the main pump will have been sized too small for the system.

Reservoir sizing, construction, and sub-component details

Refer to Figure 9.48 which is a schematic representation of an auxiliary system reservoir. Reservoir size and levels as noted in Figure 9.48 must be determined at this time. Size will be a function of system flow which previously will have been defined.

The height of the reservoir should be such that in its final field location, it will provide adequate gravity return from the main equipment.

The construction of the reservoir should be checked at this time. The original equipment vendor should have a reservoir drawing that details the reservoir internals available for review. Attention is drawn to the requirement that auxiliary return fluid should not be allowed to free fall to the surface of the liquid. All returns should be through stilling tubes or sloped troughs. It is also wise to confirm that internal design is proven and that the manufacturer has successfully designed similar reservoirs in the past. Accessibility for cleaning should be confirmed and the location of return connections and pump supply nozzles should be such that maximum residence time of system fluid is assured. Material of construction should be confirmed at this point and all details of the following reservoir sub-components reviewed:

A. Reservoir heater sizing calculations

B. Level control alarm

C. Connection locations and size

D. Additional instrumentation

Pump and driver sizing

Pump performance

Regardless of the types of pumps used, centrifugal or positive displacement, the performance curves should be reviewed at this point.

A. *Positive Displacement* – Positive displacement pumps, furnished without external timing gears, are mechanically sensitive to fluid viscosity. The performance curve should be checked at all operating points to confirm that adequate rotor separation is present at low fluid viscosities. If an operating point is at the end or within 20% of a pressure vs. flow curve at a low operating viscosity (50–60 SSU), the pump vendor should be contacted to confirm a correct selection has been made. Refer to Figure 9.49 for an example of this case.

B. *Centrifugal pumps* – Since centrifugal pump performance must be corrected for viscous fluid operation, pump sizing must confirm that the actual operating points are not close to the operating extremities of the corrected curve. That is, any operating point should not be less than 20% of pump best efficiency point nor be more than 110% of best efficiency point of the operating curve corrected for viscosity. Refer to Figure 9.50 for an example. Operation outside the stated boundaries, in addition to causing high revenue costs due to lower efficiency, can jeopardize the reliable operation of the pump.

Pump mechanical requirements

Pump data sheets must be checked at this point to confirm that proper pump case material design, bearings, seals and pump flushing arrangements are provided as specified. In addition, it is recommended that all pumps be factory tested prior to the auxiliary system test to confirm acceptable operation.

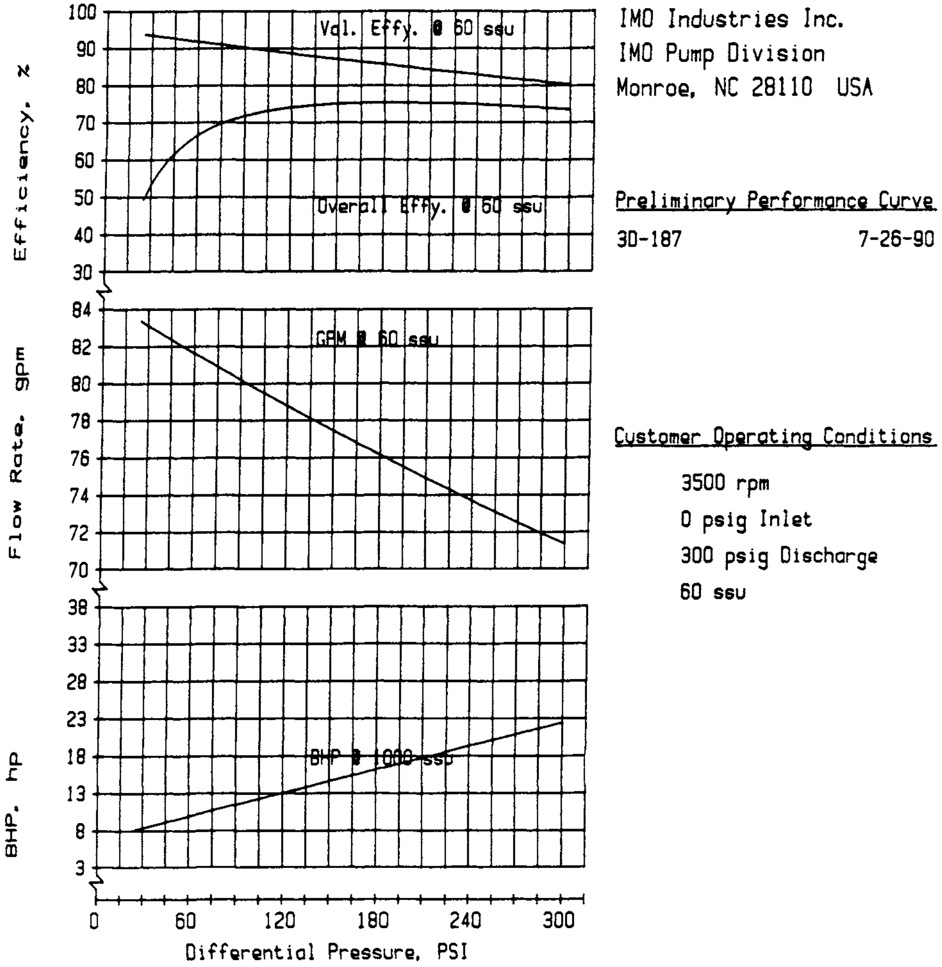


Figure 9.49 Screw pump performance (Courtesy of IMO Industries)

Pump unit couplings

Couplings should be selected for the maximum driver horsepower and include a sizing safety factor (usually 20–25%) above the maximum driver horsepower rating. Spacer couplings are recommended in order to provide ease of maintenance and minimize the necessity to remove a pump or driver while the critical equipment unit is operating. The type of coupling selected should be of high quality and reliability and provide a minimum of three years continuous operation. While either batch lube gear type couplings or dry flexible element types can be used, the latter types are preferred for their low maintenance requirements. Coupling material should be steel as opposed to cast iron

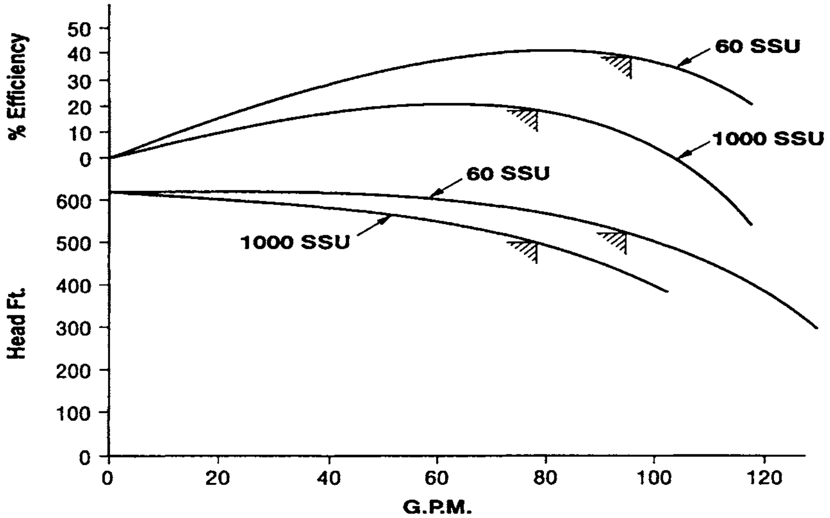


Figure 9.50 The effect of oil viscosity on centrifugal pump performance

to prevent breakage during removal or during extreme temperature changes (as during a fire). Flexible elements should be stainless steel.

The coupling shaft fit configuration and amount of shrink fit should be checked to confirm correct values.

Driver sizing

Driver sizing must be confirmed to assure adequate delivered horsepower during all operating conditions. Utility conditions to the drivers should be rechecked at this point to assure that values are as stated on data sheets. As an example, steam turbine data (inlet pressure and temperature and exhaust temperature) should be checked so that all conditions as stated will exist on site. Similarly, minimum starting voltage for motor drivers should be confirmed. Lower minimum starting voltage values than stated on data sheets will cause stand-by pump start time to be less than anticipated, shorten motor life, and could result in serious transient auxiliary pump start problems that could cause critical equipment shutdown.

Driver sizing must be confirmed with specification requirements such that driver horsepower equals pump horsepower times a specified service factor. Selection charts for expansion turbines should be checked and confirmation that proper standard size electrical drivers have been selected should be checked. In applications where viscous fluids are used, pump calculations for horsepower corrections at maximum fluid viscosity must be confirmed. Attention is drawn to realistic sizing of pumps and drivers concerning viscosity. If minimum site ambient is below 40°F for example, and a properly sized reservoir heater is furnished, there will not be a requirement for high viscosity operation if it is accepted that reservoir heater will bring the auxiliary fluid to a minimum pump starting temperature prior to pump operation. A permissive temperature switch could be installed to preclude the possibility of equipment start prior to acceptable temperature conditions.

Driver mechanical requirements

Data sheets for both main and auxiliary drivers should be checked to confirm proper mechanical design.

Motor drivers should be designed as specified with attention being paid to bearing design and motor housing design. Many smaller auxiliary systems have utilized aluminum frame motors in the past. Due to the high coefficient of thermal expansion of aluminum (double that of steel), these motors are subject to significant alignment changes with operating temperatures and could cause coupling misalignment problems.

Expansion turbine mechanical review should include governor system confirmation and safety system confirmation. Some safety valves furnished with small expansion turbines are not designed for positive shut off. This can result in operation of the turbine at lower speed once the equipment has been tripped. Most steam turbines presently operating in auxiliary systems, do not have speed indicators. To assure correct operating speed a stroboscope or hand-held tachometer, both of which can give inaccurate readings, are used. Particularly in the case of dynamic pumps, turbine speed setting is important to assure proper flow to the system. Therefore, any new installation incorporating expansion turbines should be equipped with speed indication.

Relief valve selection

Relief valve selection should be confirmed to qualify proper size, minimum accumulation (the pressure required over the valve setting to provide full flow) and chatter free operation. Relief valves should be located as close as possible to the pump discharge line to minimize the possibility of air entrainment in the line to the relief valve which can result in a delayed pump flow to the unit. This would be the case if the RV's were mounted on the reservoir a significant distance from the pump discharge line.

Control valve selection

Control valve data sheets for each control valve in the system should be available for review. Information furnished on control valve data sheets should be complete in terms of valve sizing, actuator selection and valve controller (if furnished).

Valve C_V – All operating valve coefficients (C_V's) should be stated on the control valve data sheet. That is, the normal C_V, maximum C_V and minimum C_V. These values should be compared with the selected valve internals to assure that all operating conditions fall within 10% to 90% of the maximum valve coefficient. Failure to confirm this can lead to valve instabilities. When reviewing valve coefficients, the system design must be reviewed (system schematic) since certain changes in the system could render the valve unstable.

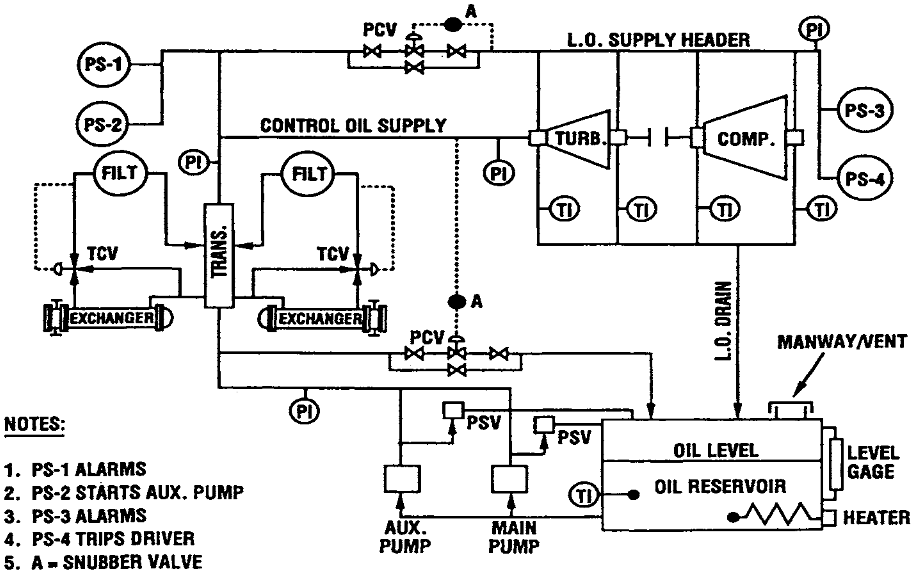
Bypass valve

For this application, the valve back pressure is atmospheric and the control valve differential depends on the condition of the auxiliary system cleanliness and any additional control valve setting (refer to the typical system schematic Figure 9.51).

C_v Minimum – The minimum valve coefficient in this application would be with a dirty filter (filer ΔP) and one pump operating.

C_v Maximum Normal – The normal valve coefficient in this application occurs with a clean system (minimum filter ΔP) and one pump operating.

C_v Maximum – The maximum valve coefficient would be with the main and auxiliary pumps operating and the minimum pressure drop across


PCV
A
L.O. SUPPLY HEADER
PI
PS-1
PS-2
CONTROL OIL SUPPLY
TURB.
COMP.
PS-3
PS-4
FILT
PI
FILT
PI
TI
TI
TI
TI
TCV
TRANS.
TCV
EXCHANGER
EXCHANGER
A
L.O. DRAIN
PCV
MANWAY/VENT
PI
PSV
PSV
OIL LEVEL
LEVEL GAGE
TI
OIL RESERVOIR
HEATER
AUX. PUMP
MAIN PUMP

NOTES:

1. PS-1 ALARMS
2. PS-2 STARTS AUX. PUMP
3. PS-3 ALARMS
4. PS-4 TRIPS DRIVER
5. A = SNUBBER VALVE

Figure 9.51 Typical lube oil supply system (Courtesy of ME Crane Consultant)

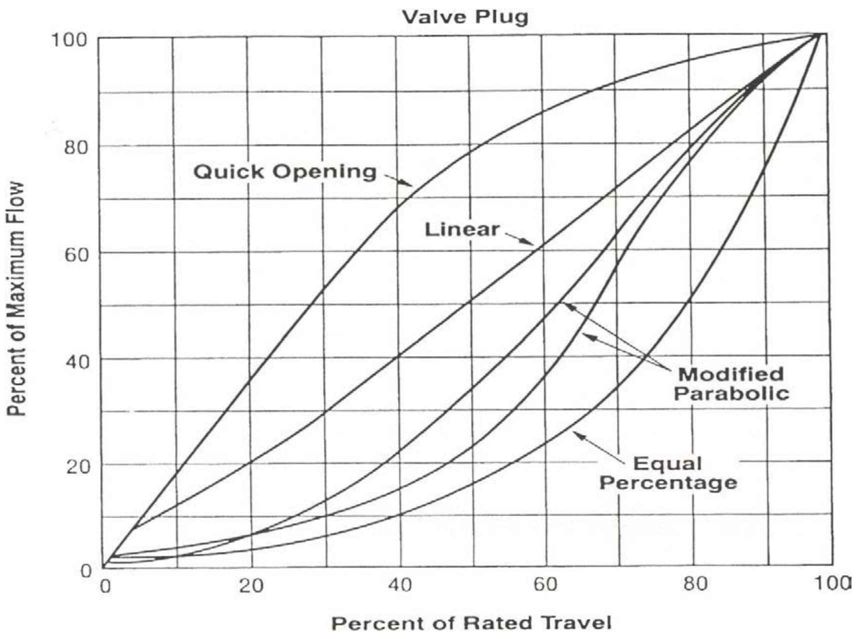


Figure 9.52 Control valve flow characteristics (Courtesy of Fisher Controls, Inc.)

the valve (clean filter). The maximum flow for this condition would be the normal bypass flow of the main pump plus the total flow of the auxiliary pump.

Attention is drawn to determining the characteristic of the valve curve for this application. The normal operating point would be approximately the minimum C_V therefore, a valve characteristic that results in a fairly significant (15–25%) valve travel for this small C_V would be desired (quick opening). Two pump operation (maximum C_V) is an abnormal case. Therefore the valve should be designed merely to pass this flow (refer to Figure 9.52) at 90% of the valve catalogue C_V.

Pressure reducing valve

In a centrifugal pump application, pressure reducing valves would experience minimum, normal and maximum C_V's similar to bypass valves with the exception that downstream valve pressure will change with increasing flow.

When pressure reducing valves are used to reduce pressure levels (control oil pressure to lube oil pressure, seal oil pressure to lube oil pressure, etc.) the valve C_V should be selected for all possible operating cases as mentioned above for bypass valves. Care should be taken to assure *all possible operating cases* are considered.

Temperature control valves

The temperature control valve C_v will remain relatively constant under all auxiliary system conditions. In the case of a two way valve however, the valve must be sufficiently sized such that the full flow pressure drop across the valve is less than the clean pressure drop across the cooler in parallel with this valve.

Differential pressure control valves and level control valves are sized and examined in the same manner as described above for bypass and pressure reducing valves. Viscosity corrections are required for all control valve sizing when operating viscosities exceed 50 Sabolt Universal Seconds (50 SSU). Significant size increases are required for high viscosity operation approaching 1,000 SSU on the order of 1½ to 2 times the selected valve coefficient without viscosity considerations.

Control valve sensing line snubber devices (dampers)

If these devices are furnished, a review of device design and confirmation of proper installation should be confirmed. Such devices provide unrestricted flow in one direction and restricted flow in another direction. The total auxiliary system operation must be reviewed in this light to confirm proper installation and orientation.

Supply pipe velocity checks

The pump header, interconnecting console pipe, and piping to the unit should all be checked for proper fluid velocity. Typical velocity values in auxiliary system supply pipes are on the order of four to six feet per second velocity. Velocity is derived from the following equation for incompressible flow.

Q ft/min = AXV

where: A = Internal pipe area (ft^2)
V = Fluid velocity (ft/sec)

Charts for standard pipe sizes and schedules are available to determine velocities (see Figure 9.53). Note that schedule 80 is usually used for carbon steel pipe below 2". Schedule 40 is used above 2". For stainless steel pipe, schedules 10 and 20 are used respectively.

Typical drain line velocities are ½ to ¼ feet per second. Attention is drawn to properly sizing drain pipes for installations where critical equipment is significantly elevated above reservoir. All drain pipes should be sized with adequate area to preclude excessive air being entrained with the oil to promote drainage back to the reservoir. An additional consideration for supply headers at the unit is that supply headers are frequently sized for one standard pipe dimension. In the

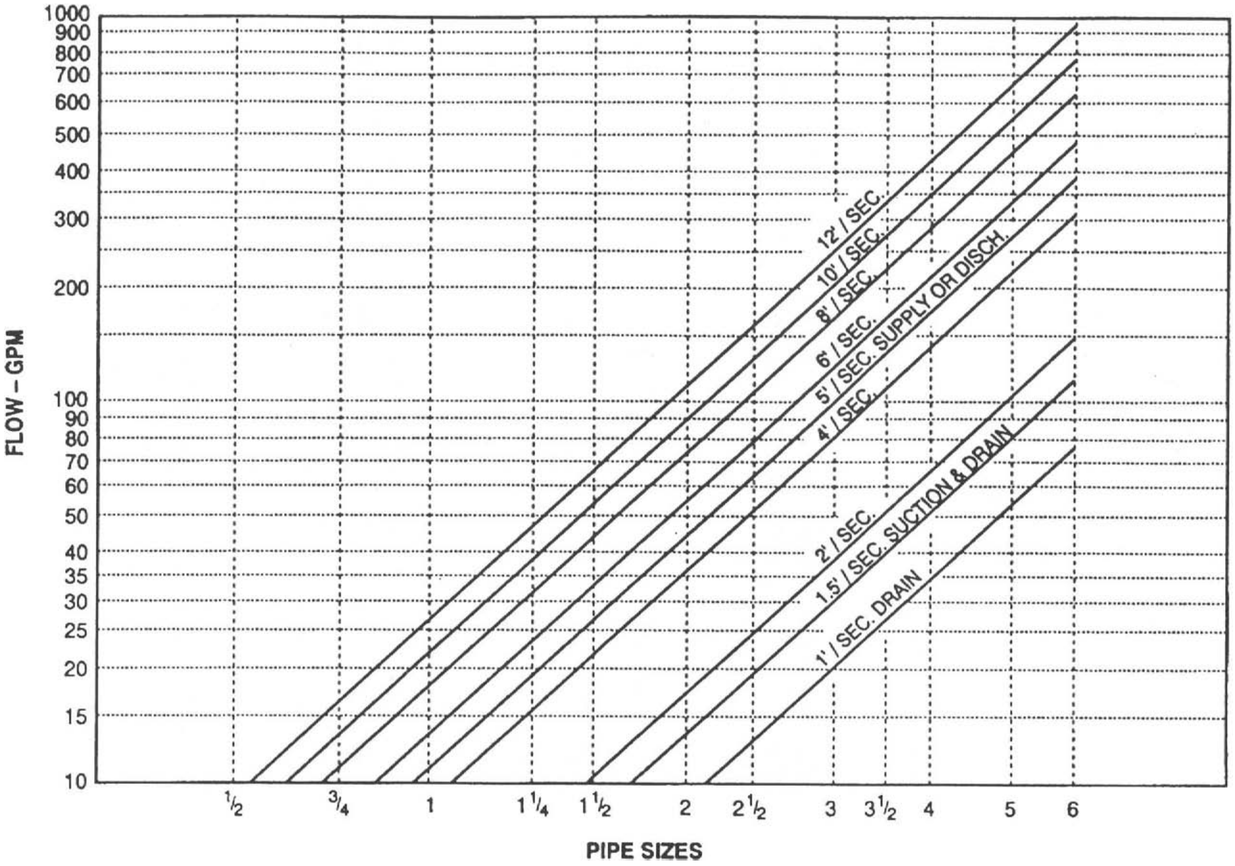


Figure 9.53 Typical pipe sizing chart

case of large critical equipment units (two or three bodies and driver), the amount of oil from the entrance to the header to the last component decreases significantly. In an effort to minimize pipe size, many vendors size headers small. Therefore, pressure drop in the header is excessive and requires a higher supply header pressure than anticipated in the unit design. Improper sizing of critical equipment supply headers could cause excessive flow across equivalent orifices (bearings) thereby requiring all flow of the main pump and necessitating the operation of the auxiliary pump.

Transfer valve sizing

Transfer valve configuration and materials of construction should be confirmed at this point. Transfer valve design should be checked to confirm tight shutoff.

Cooler sizing

The cooler data sheet should be reviewed to confirm correct duty, confirmation of correct of cooling media details, fouling factors and materials of construction.

Filter sizing

Filter information should be reviewed to confirm correct filter sizing for the normal and the maximum viscosity case (in the case of viscous fluids). Additionally, maximum filter collapse pressure, internal filter cartridge design and cartridge sealing design should also be reviewed at this time.

Instrumentation

All instrumentation should be reviewed to confirm proper selection, materials of construction and proposed installation locations. All instrumentation loops should be reviewed to assure that critical instrumentation can be calibrated and maintained while unit is in operation.

Console layout and component arrangement

Having confirmed the acceptable component sizing and selection, the console layout and arrangement of components must be reviewed. Methods of this review incorporate either the review of outline drawings of the proposed arrangement or CAD 3D drawings, a model review or a CAD 3D drawing review. Many vendors and users have found that models aid greatly in understanding and reviewing maintenance accessibility and layout considerations.

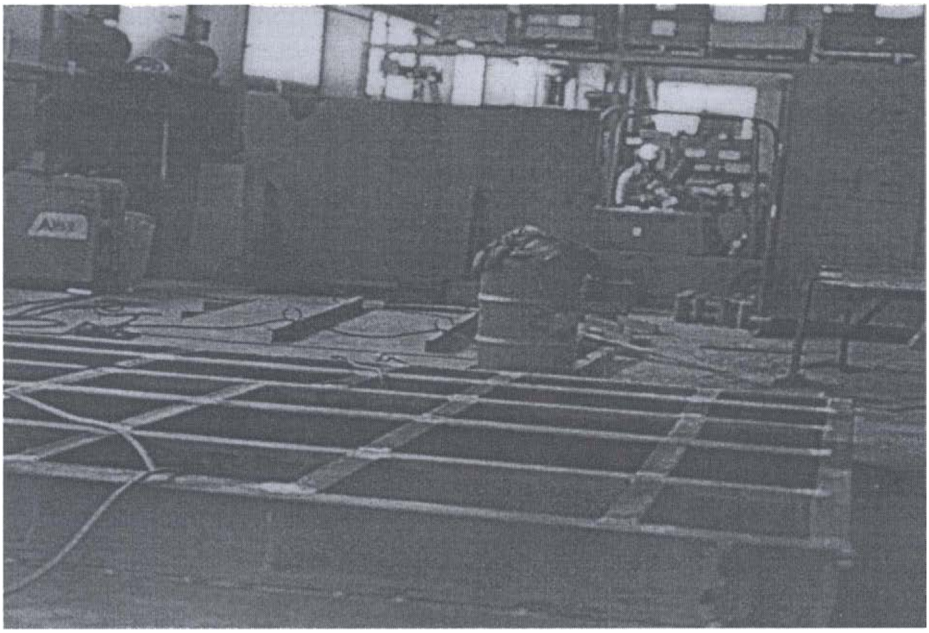


Figure 9.54 Console baseplate construction (Courtesy of Fluid Systems)

Console construction

Auxiliary equipment consoles or modules house most of the components present in the auxiliary systems. Their construction should be reviewed to assure proper stiffness and facilities for installation on site. Many horizontal consoles are constructed in a flexible manner that can result in bending or excessive pipe strains introduced into components during shipment and at installation. It is suggested that full length cross members be positioned as a minimum under pumps, coolers and filters on the equipment baseplate (see Figure 9.54). If the baseplate is to be grouted in the field, grout and vent holes should be specified and reviewed for accessibility to pore grout when equipment is installed on the baseplate.

Maintenance accessibility

Since equipment must be maintained and calibrated while the auxiliary system is in operation, it is important to provide ample personnel space such that equipment can be maintained safely and reliably without damage to surrounding components. A rule of thumb is to provide approximately one meter of space around components for accessibility. Note that this is with the utility lines installed. The review of equipment on a model CAD 3D drawing or an outline should be made considering installation of all utility lines that will be installed in the field.

On-line testing and calibration accessibility

Considering that many components (pumps, drivers, coolers, filters, control valves, instrumentation) will be tested and calibrated with equipment in operation, accessibility for this operation must be considered.

In addition to reviewing the vendor manufactured skids, the placement of all skids in the field must be reviewed for accessibility. Consideration of the skid arrangement only to be complicated by installation against a column or wall in the field will not obtain the objectives of total accessibility.

Utility supply arrangement

Care should be given to the routing of all utility (conduits, steam lines, water lines) supply lines in order to maximize accessibility to the critical equipment auxiliary systems.

Considerations for component disassembly

All components must be able to be dissembled quickly, easily and safely while the unit is operating in the field. To meet this requirement, sufficient space around the auxiliary console must be available for such exercises as cooler bundle removal, filter cartridge removal and auxiliary or main driver removal. In addition, consoles are frequently installed in congested areas and lifting arrangements should be reviewed beforehand to confirm components can be removed in a safe and easy manner.

This completes comments concerning the component sizing audit. All changes made during this meeting should be documented and followed up to guarantee that final component design and arrangements are as specified and agreed to in this meeting.

Auxiliary system component sizing audit form

Applicable specifications

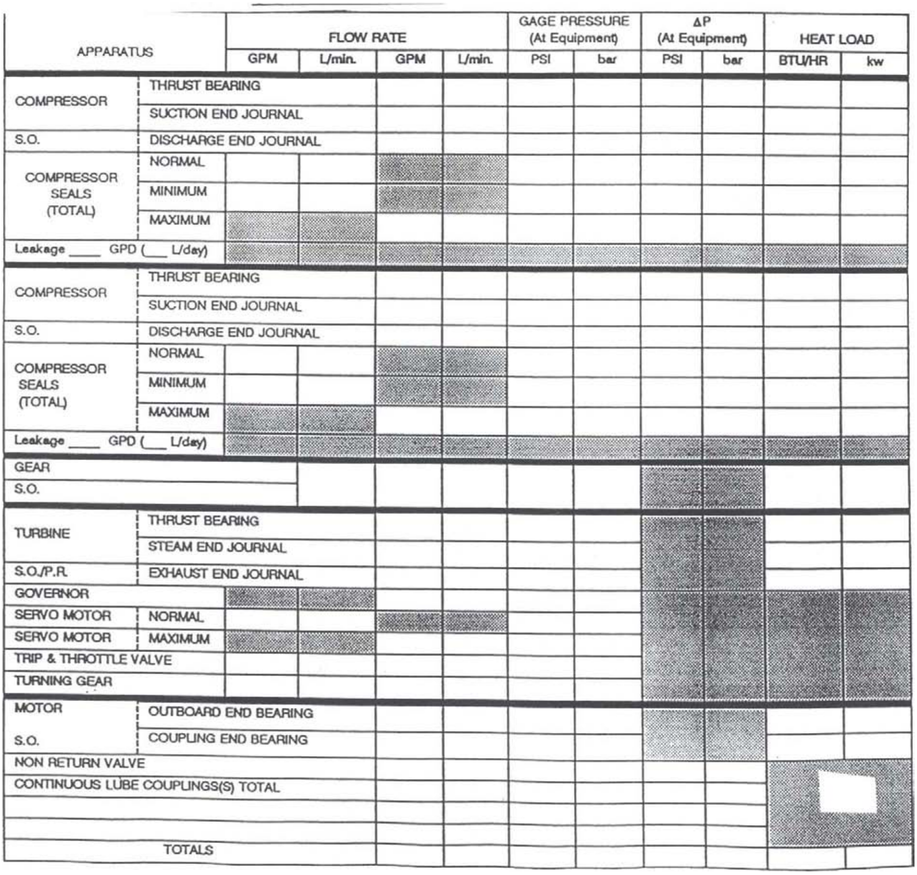
Critical equipment description

Schematic drawing numbers

Data sheet

I. System requirements

A. Oil requirements



APPARATUS		FLOW RATE				GAGE PRESSURE (At Equipment)		ΔP (At Equipment)		HEAT LOAD	
		GPM	L/min.	GPM	L/min.	PSI	bar	PSI	bar	BTU/HR	kw
COMPRESSOR	THRUST BEARING										
	SUCTION END JOURNAL										
S.O.	DISCHARGE END JOURNAL										
COMPRESSOR SEALS (TOTAL)	NORMAL			▨	▨						
	MINIMUM			▨	▨						
	MAXIMUM	▨	▨								
Leakage ____ GPD (___ L/day)		▨	▨	▨	▨	▨	▨	▨	▨	▨	▨
COMPRESSOR	THRUST BEARING										
	SUCTION END JOURNAL										
S.O.	DISCHARGE END JOURNAL										
COMPRESSOR SEALS (TOTAL)	NORMAL			▨	▨						
	MINIMUM			▨	▨						
	MAXIMUM	▨	▨								
Leakage ____ GPD (___ L/day)		▨	▨	▨	▨	▨	▨	▨	▨	▨	▨
GEAR								▨	▨		
S.O.								▨	▨		
TURBINE	THRUST BEARING							▨	▨		
	STEAM END JOURNAL							▨	▨		
S.O./P.R.	EXHAUST END JOURNAL							▨	▨		
GOVERNOR		▨	▨					▨	▨	▨	▨
SERVO MOTOR	NORMAL			▨	▨			▨	▨	▨	▨
SERVO MOTOR	MAXIMUM	▨	▨					▨	▨	▨	▨
TRIP & THROTTLE VALVE								▨	▨	▨	▨
TURNING GEAR								▨	▨	▨	▨
MOTOR	OUTBOARD END BEARING							▨	▨		
S.O.	COUPLING END BEARING							▨	▨		
NON RETURN VALVE										▨	▨
CONTINUOUS LUBE COUPLINGS(S) TOTAL										▨	▨
										▨	▨
										▨	▨
TOTALS											

Figure 9.55 Auxiliary system component sizing audit form

I.B. System requirements

1. Total pump flow (Positive displacement pump) = ________ x equipment flow (I_a)
   = ________

   Total pump flow (Centrifugal pump) = ________ (total flow in I_a)

2. Bypass flow (positive displacement pump) = (1) – I_a total flow
   = ________ - ________
   = ________________

3. Total head load (from I_a) = ____________ BTU/HR

4. Pump discharge pressure

Viscosity (SSU)	60	1000
A. Lube oil pressure (at equipment)	______	______
B. Elevation ΔP	______	______
C. Pipe ΔP	______	______
D. Valve ΔP	______	______
E. Cooler ΔP	______	______
F. Filter (clean) ΔP	______	______
G. Miscellaneous ΔP	______	______

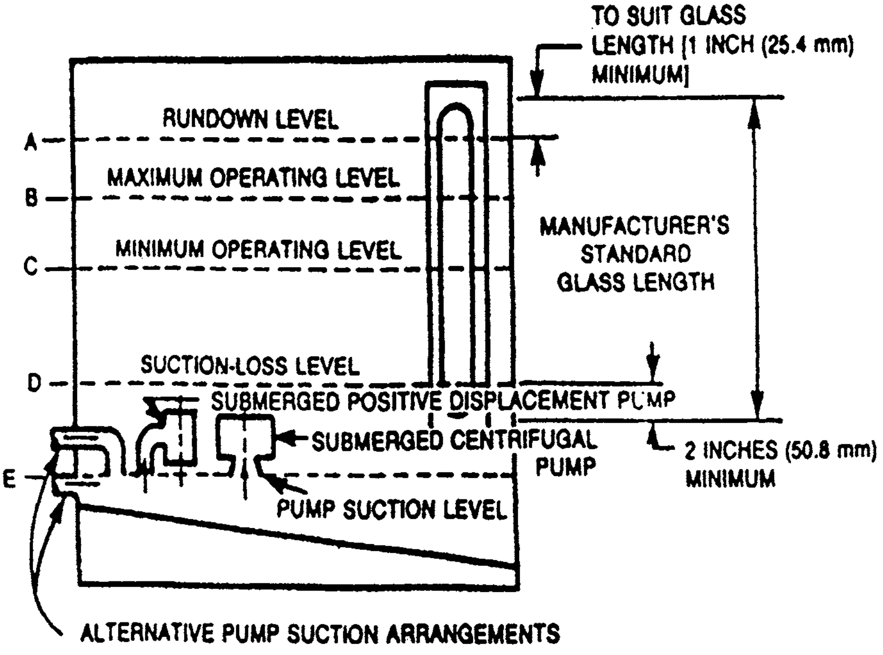

TO SUIT GLASS LENGTH [1 INCH (25.4 mm) MINIMUM]
RUNDOWN LEVEL
A
MAXIMUM OPERATING LEVEL
B
MINIMUM OPERATING LEVEL
C
MANUFACTURER'S STANDARD GLASS LENGTH
SUCTION-LOSS LEVEL
D
SUBMERGED POSITIVE DISPLACEMENT PUMP
SUBMERGED CENTRIFUGAL PUMP
2 INCHES (50.8 mm) MINIMUM
E
PUMP SUCTION LEVEL
ALTERNATIVE PUMP SUCTION ARRANGEMENTS

Figure 9.56 Reservoir levels and oil level glass details

Pump discharge press ______PSI ______PSI
(Add A through G)

Note: If system is combined with seal and/or control oil, add highest value to determine pump discharge pressure.

II Component requirements

Confirm sizing as started below and check data sheet and specific requirements for each component.

A. Pump selection

	Positive displacement main/auxiliary	Centrifugal main/auxiliary
1. Pump type	______	______
2. Make	______	______
3. Model	______	______
4. *Speed	______	______
5. Disch. Press @ 60 SSU (rated)	______	______
6. Disch. Press @ Max SSU	______	______
7. Rated flow @ 60 SSU	______	______
8. Flow @ Max SSU	______	______
9. Flow @ relief valve pres (positive displacement pump only)	______	______
10. Rated BHP	______	______
11. Max. BHP (@ R.V. and max. viscosity)	______	______
12. NPSH available	______	______
13. NPSH required	______	______
14. Suction lift (if pumps are mounted above fluid level)	______	______

*If steam turbine driver is used, it is recommended that speed should be 2 pole (3600/3000 RPM) motor speed to minimize steam rate.

B. Coupling selection

	Main	Auxiliary
1. Pump	______	______
2. Coupling model	______	______
3. Size	______	______
4. Driver max. HP	______	______
5. HP/100 RPM	______	______
6. Coupling HP/100 RPM	______	______

Note: Confirm appropriate coupling service factor is used. Rotary (P.D.) pumps require a higher service factor.

C. Driver selection

	Main	Auxiliary
1. Service	______	______
2. Type	______	______
3. Speed	______	______
4. Pump max. HP	______	______
5. Driver rated HP (= 1.1 x pump max. HP)	______	______
6. Driver normal HP (@60 SSU)	______	______
7. Turbine steam rate (max/rated)	______	______
8. Steam quantity @ minimum steam energy condition LB/HR	______	______
*9. Driver starting time 0 RPM-rated speed	______	______

Note: Confirm sufficient steam is available at minimum energy conditions.

*Calculated for minimum energy conditions (minimum steam energy or motor minimum starting voltage) and pump rated conditions. If greater than three (3) seconds, accumulator(s) should be used.

D. Relief valve selection (Positive displacement pumps only)

1. Pump max. discharge pressure at max. viscosity = ______

2. Relief valve pressure = 1.1 x D1 or 25 PSI. greater, whichever is higher

Relief valve type (modulating preferred) = ______

Model = ______

Set pressure = ______

Overpressure (pressure to pass full flow) = ______

Normal leakage (valve closed) = ______

E. Reservoir sizing (based on rectangular tank) per API 614

1. Normal flow (GPM) = ______

2. Retention time (minutes) = ______

2A Capacity = (1) x (2)

Confirm size

3. Reservoir length and width (inches) = ______ , ______

4. Capacity/inch of height $= \dfrac{(3)}{231 \text{ in.}^3/\text{gal}}$

= ______ gal/inch

5. Level E (pump suction level) = ________ in. above grade
6. Level D (suction loss level) = (5) + level required to maintain prime
   = ________
7. Level C = (6) + 5 (minutes) x(1)

   Note: 5 minutes = working time

   Working capacity (volume between levels C & D) *or* level C =

   $$\frac{8\,(\text{min}) \times (1)}{(4)} + \text{tank bottom height above grade}$$

   Note: 8 minutes = retention time

   Retention capacity = volume between bottom of tank and level C.
8. Level B = Highest level of oil during operation (approximately 1 minute retention time)
9. Level A = Highest level oil can reach
   = Level B + capacity contained in all components that drain back to the reservoir ÷ (4)

   Note: This quantity should also include allowance for interconnecting piping and any overhead tanks.
10. Minimum reservoir free surface area:

    = 0.25 $ft.^2$/gpm of normal flow

    = 0.25 x (1)
    = ________ ft^2

    Confirm reservoir internals, material, etc. meet data sheet and specifications required. Review reservoir internal drawing.

F. Reservoir heating requirements

Type: Electric Steam

Requirements

Time to head oil from ________ °F to ________ °F = ________ hours

1. Calculated heat load = _________ BTU's
   (minus reservoir heat loss)

2. Heater size BTU/HR = $\frac{\text{(F1)}}{\text{Total time allowed (hours)}}$

3. Electric heater max. watt density = ________

   Note: Confirm if heaters can be removed without draining reservoir.

G. Supply pipe velocity

Maximum velocity 4–6 ft/sec.

Maximum console supply pipe velocity = ________________ft/sec.

Maximum unit supply pipe velocity = ________________ft/sec.

H. Control valve sizing

H1. Bypass (back pressure) valve

1.1 Type: self acting, pneumatic or electric controller ____________

1.2 Make ____________

1.3 Model ____________

1.4 Action – Direct or Reverse ____________

1.5 Valve plug type ____________

1.6 Failure mode ____________

1.7 Actuator size ____________

1.8 Actuator force available/force required ____________

1.9 Maximum valve C_V = ____________

1.10 Operating C_V min. (one pump dirty system) = ____________

1.11 Operating C_V max. (two pumps clean system) = ____________

Note: Operating C_V's should be between 10% – 90% of valve max. C_V.

Sensing line pulsation snubber required? If so, confirm proper orientation. Confirm fast response is to *open or close* valve.

H2. Transfer valve (s)

2.1 Make ____________
2.2 Model ____________
2.3 Size ____________
2.4 Plus type – taper, straight, globe ____________
2.5 Lifting jack required? ____________
2.6 Tight shut-off required? ____________
2.7 Max ΔP on changeover ____________

H3. Temperature control valve(s)

3.1 Make ____________
3.2 Model ____________
3.3 Size ____________
3.4 Normal flow ____________
3.5 Temperature range ____________
3.6 Valve max operating C_v ____________
(if 2-way valve, C_v must be based on clean cooler)
3.7 Valve maximum C_v ____________

Note: Butterfly type valve often used for 2-way applications.

H4. Pressure reducing valve

4.1 Type: self acting, pneumatic or electric ____________
4.2 Make ____________
4.3 Model ____________
4.4 Action-direct or reverse ____________
4.5 Valve plug type ____________
4.6 Failure mode ____________
4.7 Actuator size ____________
4.8 Actuator force ____________
4.9 Maximum valve C_v = ____________
4.10 Normal valve operating C_v = ____________
(unit at operating speed)
4.11 Minimum valve operating C_v = ____________
(unit at rest – oil system on)

I. Cooler sizing

Type	Shell and tube	Air (fin fan)
1. Twin or single	________	________
2. Make	________	________
3. Model	________	________
4. Size	________	________
5. Heat load btu/hr	________	________
6. Oil side ΔP clean (psi)	________	________
7. Fouling factor (total)	________	________
8. Oil flow (gpm)	________	________
9. Water quantity (gpm)	________	

J. Filter sizing

1. Make ________
2. Model ________
3. Type – surface ________ depth ________
4. Normal flow (gpm) ________
5. Max. flow (gpm) ________
6. Filtration (microns) ________
7. Clear filter ΔP max. ________
8. Cartridge material ________
9. Type end seals ________
10. Cartridge-single or multiple ________
11. Cartridge center tube material ________
12. ΔP at max. viscosity ________
13. Collapse pressure ________
14. Number of cartridges ________
15. Gpm per cartridge ________

K. Switches or transmitters

Confirm proper range, type, materials and maximum deadband (change in actuation point) of each switch. Confirm proper selection of transmitters.

L. Gauges

Confirm proper range, type, material of each pressure, differential pressure, temperature and level gauges.

M. Accumulator sizing

1. Type: bladder _______ or direct acting _______
2. System flow (gpm) = _______
3. System transient time (sec.) = _______
4. Capacity of fluid required = $\frac{(2) \times (3)}{60}$ = _______ Gallons
5. System pressure below which accumulator begins to drain (PSIA) = _______
6. Precharge pressure (PSIA) _______
7. Proposed accumulator internal volume (approximately 90% of normal size) = _______
8. Actual fluid capacity per accumulator = _______

   $$(7) \times \left[1 - \left(\frac{(6)}{(5)}\right)\right] = \_\_\_\_\_\_\_$$

9. Precharge type: manual, self contained, automatic _______

N. Additional tank sizing and construction confirmation

Overhead rundown (lube)

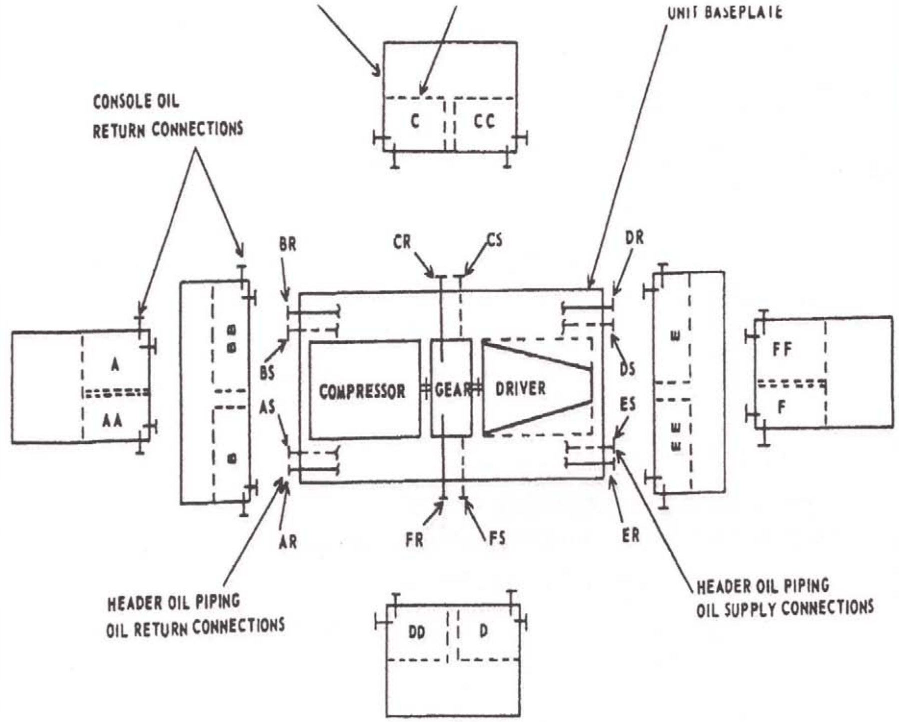
Overhead (seal)

Degassing tank(s)

These tanks should be checked against specifications data sheets for proper capacity, construction and ancillaries.

O. Piping, vessel, flange and component material

Refer to Figure 9.57 and finalize all connection locations.


UNIT BASEPLATE
CONSOLE OIL RETURN CONNECTIONS
C
CC
BR
CR
CS
DR
A
AA
BB
B
BS
AS
COMPRESSOR
GEAR
DRIVER
DS
ES
E
EE
FF
F
AR
FR
FS
ER
HEADER OIL PIPING OIL RETURN CONNECTIONS
HEADER OIL PIPING OIL SUPPLY CONNECTIONS
DD
D

AQ. Standard console location relative to unit (identify by letter)

Normal arrangements	☐A	☐B	☐C	☐D	☐E	☐F
Inverted arrangements	☐AA	☐BB	☐CC	☐DD	☐EE	☐FF
Header oil piping supply oil conn.	☐AS	☐BS	☐CS	☐DS	☐ES	☐FS
Header oil piping oil return conn.	☐AR	☐BR	☐CR	☐DR	☐ER	☐FR
Conns. to terminate at	☐ Edge of unit baseplate* ☐________________					

Figure 9.57 Connection orientation drawing (Courtesy of Elliott Company)

Management awareness workshop agenda example

Day 1

Session	Topic
1	Workshop introduction and overview ■ Objectives ■ Schedule ■ Agenda review ■ The benefits of improved reliability to machinists and operators ■ How machinists and operators can contribute to improved reliability ■ Important concepts (reliability, MTBF, machinery environment, bad actors)
2, 3, 4, 5, 6, 7	The root causes of machinery failures (the 5 why's) ■ Introduction ■ Process condition changes ● Pumps ● Steam turbines ● Auxiliary system problems ■ Installation errors ■ Design problems ■ Operating procedures ● Pumps ● Steam turbines ■ Component wearout ■ Auxiliary system problems
8	How to prevent machinery failures ■ Component function awareness ■ Component condition monitoring ■ Predictive maintenance techniques ■ Reliability is everyone's responsibility

Day 2

9, 10 Component condition monitoring and predictive maintenance

- The major components in any machine
- Rotor condition monitoring
  - Pumps
  - Steam turbines
- Journal bearing condition monitoring
- Thrust bearing condition monitoring
- Seal condition monitoring
  - Pumps
  - Steam turbines

11, 12 Steam turbine reliability factors (effective warm-up, cool down procedures)

- Procedures
- Monitoring

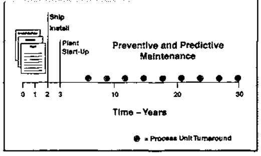
13, 14 Centrifugal pump reliability factors (hydraulic disturbances – cavitation, re-circulation)

- Causes
- Determination of root cause
- Prevention

15, 16 Workshop summary

- How 'learned principles' can be implemented
- Action plan by attendees

10


Ship
Install
Plant Start-Up
Preventive and Predictive Maintenance
0 1 2 3
10
20
30
Time – Years
● = Process Unit Turnaround

Rotating equipment reliability assurance

- Introduction
- The pre-FEED phase
- The specification and ITB phase
- Pre-bid activity and degree of audits
- Bid evaluations
- Pre-award meeting
- The coordination meeting
- Design and manufacturing audits
- Document review
- Testing phase

Appendix
- Suggested vendor pre-bid meeting details letter
- Pre-bid procedure fact summary
- Typical compressor train pre-bid meeting agenda
- Typical FAI compressor bid tabulation
- Shop test checklist

Introduction

As someone who has been involved with projects as a rotating equipment vendor, end user and consultant since 1970, I have had the opportunity to see custom designed rotating equipment projects from all industry viewpoints. Regardless of your position in this subject, you will face the challenges of company profit optimization, depleted workforce experience levels and time constraints.

My initial involvement with rotating equipment projects began in 1970 as a project engineer for a centrifugal compressor vendor where I was responsible for project management of all process compressor applications. This interesting and busy portion of my career taught me many valuable lessons and the challenges and associated action required to survive this experience. 'Vendor lessons learned' are detailed in Figure 10.1.

Vendor lessons learned

- Time constraints forced the acceptance of what was on the data sheet
- The tendency was to think inside the flanges of the compressor only and not consider the process
- Questions to the end user/contractor were minimal based upon competitive pressures and time constraints
- Copying from past jobs 'cut and paste' was a necessity to minimize engineering hours
- Contractor/end user questions diminished valuable engineering time
- There was little time or money for visits to client plants unless there were significant design problems

Figure 10.1 Vendor lessons learned

It was interesting to note that in my next industry position as a corporate rotating equipment specialist for a major oil, gas and chemical company, I observed that the characteristics in Figure 10.1 were present in all equipment companies regardless of global location or final product. However, in my new position there were also many challenges as noted in Figure 10.2.

End user lessons learned

- Time constraints forced acceptance of what was on the process data sheet without time to question the basis for the stated conditions
- The tendency initially was to think inside the machinery flanges, but eventually it was understood that all equipment is directly influenced by the process
- Contact with the client (plant where the equipment will be installed) was minimal based on project team pressures for schedule milestones
- Company specification contents were increasing rapidly since all company divisions and plants were required to review specifications and therefore naturally contribute something
- There was limited project budget for visits to client plants unless there were equipment design problems.

Figure 10.2 End user lessons learned

Review Figures 10.1 and 10.2 and observe the similarities all imposed by time and budget constraints. Also observe how the involved individuals seldom have the opportunity to observe how their client operates and what his objectives are.

Since 1990, as a rotating equipment consultant engaged in troubleshooting, machinery selection and revamps and site specific operator, maintenance and engineering training, I have other challenges but the similarities are striking and the challenges are the same. These facts are noted in Figure 10.3.

Contractor/consultant lessons learned

- Both vendors and clients have limited experience bases
- Decisions are made quickly, often without benefit of all the pertinent facts
- Most projects are run on the basis of minimum capital investment and not life cycle cost
- Implementation of action plans is slow
- Vendor and end user's interface infrequently – usually only during field failures

Figure 10.3 Contractor/consultant lessons learned

Based on my experience, I have learned, most of the time the hard way, that all three of these groups (vendors, contractors and end users) have the same objective but different means of obtaining that objective. Figure 10.4 presents these facts.

The objective – maximum profits

Everyone has the objective of maximum profits but the means to accomplish this end is different:

- Vendor – designs for minimum cost
- Contractor – engineers and installs for minimum cost
- End user – must operate the custom designed equipment 24/7 for 30 years

Therefore, the end users objectives can be directly opposed to the vendor's and contractor's!!!

Figure 10.4 The objective – maximum profits

It is important to remember these facts at all times during the entire project. The information contained in this figure should be the basis for convincing the project team that all decisions regarding equipment purchase should be made on the basis of life cycle cost and not capital cost and/or schedule considerations. The specific objectives of the end user are presented in Figure 10.5.

End user – specific objectives for maximum profit

- Maximum machine reliability
- Minimum operating cost
- Minimum time to repair

These objectives result in.. maximum up time

which will yield .... maximum revenue

and .... maximum profits

over the entire life cycle of the process unit!!

Figure 10.5 End user – specific objectives for maximum profit

The most important factor in life cycle cost considerations is daily revenue and obtaining this figure should be the number one priority in the early stages of the project. It will be a key fact in obtaining management support for your project action plans. Figure 10.6 presents these facts.

Daily revenue

- Is the amount of revenue obtained in 24 hours of operation
- Trip of an unspared item = exposure to revenue loss
- Daily revenue values can range from 1MM$ to 5 MM$+
- Always justify audit requirements and shop test scope on the basis of daily revenue loss

Figure 10.6 Daily revenue

Therefore, the company life cycle revenue and profit, potential for over 100 MM$ added profit, will be a result of incorporating all of your project best practice requirements into the project action plan at the first opportunity before the first project budget estimate is prepared. Figure 10.7 shows the advantages of incorporating this philosophy as early as possible into the project.

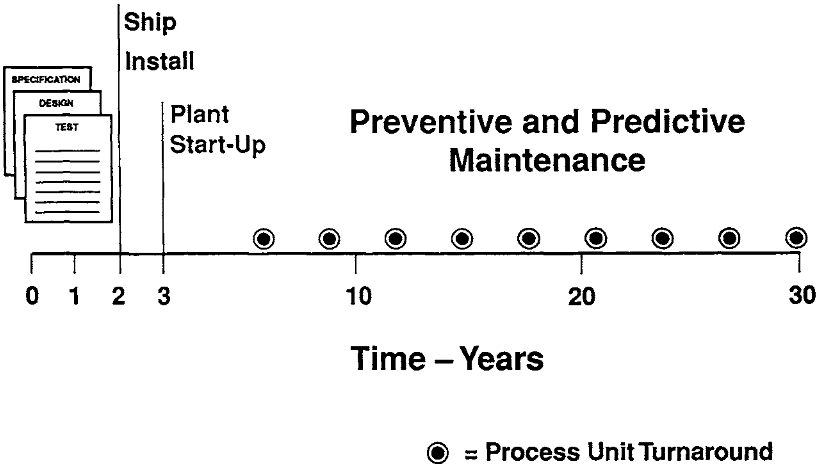

Ship
Install
SPECIFICATION
DESIGN
TEST
Plant Start-Up
Preventive and Predictive Maintenance
0
1
2
3
10
20
30
Time – Years
● = Process Unit Turnaround

Figure 10.7 The life span of rotating equipment

This action should be taken when the project is first announced and the project team is assembled. The approach taken during the first 3–6 months after the initial project kick off will determine the level of reliability and life cycle cost savings for the entire life of the process unit (over 30 years). Most important is the necessity of establishing immediate creditability with the project team so that your ideas are implemented.

Hopefully, the above information will be of use in your project involvement in terms of lessons learned. The resulting best practices should be developed into a project philosophy that will eliminate all the issues noted above and will obtain and maintain your management's support throughout the entire project from the pre-feed phase to field operation.

Before completing this section, it is important to note that while this book is concerned with rotating equipment, the principles covered in this chapter are equally applicable to all assets included in a project. Please note that in this chapter "Vendor" and "Supplier" are used interchangeably.

The pre-FEED phase

The action taken during the pre-FEED phase (Front End Engineering Design) relative to rotating equipment will set the stage for its availability and profit improvement for the entire life of the process unit. However, the company must take the initiative to assemble and brief the project team members immediately upon project inception. Corporate responsibilities are outlined in Figure 10.8.

Corporate project responsibilities

- Assemble the entire project team immediately upon project announcement
- The team should include existing plant maintenance, operations personnel or experienced personnel if the plant is a grass roots (new) installation
- Brief specialists regarding details of process, size of equipment special details etc.
- Require specialist input immediately regarding equipment special project requirements

Figure 10.8 Corporate project responsibilities

The corporate action outlined in Figure 10.8 will enable the specialist to acquire the project information that she or he needs to determine the degree of risk involved for the purchase and manufacture of the critical (custom designed) equipment on the project. In addition, the specialist can perform some preliminary selection calculations to determine what vendors can supply the equipment. Figure 10.9 presents key facts necessary to determine potential vendor capabilities.

Determine potential vendor capabilities by

- Past project experience - design errors, manufacturing problems, delays etc.
- Past field experience - availability, maintainability, field support etc.
- Vendor's reference lists for the specific project application detailing flow, head, efficiency, etc.
- Networking with industry peers (API, symposiums etc.)

Figure 10.9 Determine potential vendor capabilities

Once these facts are obtained, an acceptable vendors list for this specific project can be prepared. It is important to note that each specific project will have different requirements and some vendors will not be in the same position as their competition in terms of design experience and/or manufacturing capability.

The decision not to use a certain vendor may be difficult based on past associations but it is in the end user's interest to only select the vendors that have the most design and manufacturing experience for a specific application. Vendors not having the required level of experience for this application should be informed immediately to save them the high costs of quoting and to explain that the decision taken is not an expression of the quality of their design and manufacturing but their relative experience level for this specific application and will definitely not impact future business opportunities.

After the potential vendors are determined, the degree of risk for this application must be defined to determine the when and where audits are required. Figure 10.10 presents these considerations for critical unspared machinery.

Risk classifications and action

Risk classification	Action
Prototype	Design and manufacturing pre-screening in pre-feed phase
Multi-component inexperience	Design audit in bid phase**
Single-component inexperience	Design audit at coordination meeting**

**= Manufacturing audit may be required based on design

Figure 10.10 Risk classifications and action

After the degree of risk is determined, the specialist should review company documents and conduct discussion with plant operations, maintenance and engineering personnel to determine what special 'lessons learned' and 'best practices' are appropriate for the particular project. At this point, the specialist will be able to define the specific action plan for the equipment on this project and to prepare a project management presentation. Guidelines are presented in Figure 10.11.

Guidelines for project plan presentation

- Present envisioned equipment overview based on input data, estimation calculations, vendor discussions and best practices
- Define risk - safety and cost of un-availability
- Recommend action plan - include: audit, application best practice and special test requirements
- Define cost and additional schedule time based on recommended action plan
- Define company savings and increased profit over life of process unit
- Request decision to proceed with the proposed plan

Figure 10.11 Guidelines for project plan presentation

The format of this presentation can range from a discussion with the project engineering manager and project manager to a formal power point presentation. Regardless of the type of presentation, time is of the essence and the presentation must detail in clear and concise terms, the specific requirements, schedule time and life cycle cost savings for the proposed plan. An example of turning acquired pre-FEED information into an action plan is presented in the case history in Figure 10.12.

Refrigeration compressor case history

- Input from project team – propylene refrigeration duty data sheet
- Calculations and vendor discussions showed that duty required a prototype machine in regards to rotor bearing span and shaft diameter (shaft stiffness)
- Risk class was determined as multiple component inexperience
- Vendors were invited to design pre-screening meetings to determine action
- Based on meeting reviews with 3 vendors, it was determined that bearing span had to be reduced and that 2 compressor cases, in series, were required for proven reliability
- Costs of 2nd case were assembled along with supporting data and cost figures for exposure to reduced availability and benchmarks of problems experienced with the one case option (this information was obtained from experienced plant maintenance and operations personnel)
- The management presentation was successful and additional 5 $mm was approved for purchase and installation of 2nd compressor casing

Figure 10.12 Refrigeration compressor case history

The above action was possible because the project team provided early information to the specialists, allowed pre-screening meetings to audit vendor experience and took the specialist's recommendations seriously.

This action took place before the budget estimate. The example outlined in Figure 10.12 will become more important, in the future, as the size of projects increase and the exposure to loss of daily process unit profit can easily exceed millions of dollars.

Once the risk class requirements and best practices for the project are known and approved by project management, a philosophy can be developed to define the specific activities and schedule required for all phases of the project. This information must now be included in the ITB (invitation to bid) document in clear concise terms. Guidelines for ITB instructions are noted in Figure 10.13.

ITB instructions to vendors

- Incorporate all project team accepted items (audit, best practice and test requirements)
- Define audit details (when where and who attends)
- Define discipline and experience requirements for all participants in all scheduled meetings
- Note penalty for non-compliance (eg, bid not accepted)

Figure 10.13 ITB instructions to vendors

The specification and ITB (invitation to bid) phase

At this point, the information required for preparing the project specifications for the project should be available. The challenge will now be to format these specifications to assure complete compliance by all quoting parties and to minimize the schedule time. The selected format of the specifications will significantly affect the project schedule in terms of preparation time, vendor response time and contractor/end user/vendor time to review specification exceptions for approval. The effects of the specification format on reliability and project schedule are shown in Figure 10.14.

Specification format effects

The selected specification format can produce the following effects on equipment reliability and project schedule:

- Interpretation error - due to complexity of spec's = reliability risk and delay
- Additional preparation time - for contractor
- Additional review time - for vendors
- Additional meeting and/or document exchange time for contractor/end user review of vendor exceptions to specifications

Figure 10.14 Specification format effects

Let's face it, specifications have become very complex and extensive. It is my experience that this is due to the fact that most end users send specification drafts out to their plants and affiliates for review prior to publication and the result is that every participant has to contribute or it does not look good! The end product is a thick specification. Are specifications of this type really necessary? Another consideration regarding complex specifications is that the experience level in all phases of the industry is decreasing and the time required and accuracy of specification reviews is being affected.

Is there a viable alternative? Over the last 10 years, I have been involved with a number of medium to small companies that are beginning to build new facilities and have the flexibility to decide on specification format. Considering that their budget for specification preparation is limited, but they clearly recognize the importance of a sound specification based on industry standards and global best practices, they have generally adopted a strategy as shown in Figure 10.15.

The streamlined specification strategy

- Use established industry specifications (eg. API or ANSI)
- Include industry data sheets that are completed to outline all details of scope of supply and company lessons learned
- Attach a best practice list applicable for the specific project only
- Use a global consultant firm to review and comment
- Note in the cover letter to the specification package that strict compliance with all requirements is mandatory – options are not acceptable

Figure 10.15 The streamlined specification strategy

The approach noted in Figure 10.15 will only be possible if specialist creditability with project management has been secured and maintained. If there is a reluctance to depart from the established specification format, a recommendation would be to contact other associates who you have met at industry conferences or search the web for consultants who have experience in this regard and can support your effort. Believe me, there are many benefits to this approach for all parties (vendors, contractors and end users). Benefits to the streamlined specification approach are shown in Figure 10.16.

Streamlined specification format – benefits

- End user – low probability of misinterpretation, shorter vendor bidding cycle time and reduced review time for vendor exceptions resulting in lower cost, reduced project schedule and manpower
- Contractor – less preparation and review time resulting in lower cost, reduced project schedule and manpower
- Vendor – less review and exception preparation time resulting in lower cost, reduced project schedule and manpower

Figure 10.16 Streamlined specification format – benefits

The streamlined specification approach as shown in Figure 10.15 is really a win win for all participants but for larger, established companies, it definitely is a culture change and will have its price in terms of learning curve and initial expended revenue.

As mentioned, any new or different approach will meet initial resistance (a 'paradigm shift') and will require specific 'instructions to the bidder' in clear, concise and brief terms. Figure 10.17 presents some guidelines.

ITB guidelines

- Use 'executive format' style and list specific requirements in the first page of the ITB
- Examples of requirements: pre-bid meeting, audit, exception to specification rule details
- Require written acknowledgement of compliance immediately from all vendors

Figure 10.17 ITB guidelines

An example of a pre-bid meeting letter used in an ITB is included at the end of this chapter. Note that this letter can easily be modified for audit requirements, which are discussed in detail in a later section of this chapter. It has been my experience that the guidelines presented above for a streamlined approach, if endorsed by the project management team and implemented as noted, can result in considerable schedule savings and increased equipment reliability. Because of its importance in saving project schedule time, I have repeated an important ITB message to vendors in Figure 10.18.

Important ITB message to all vendors

- The consideration given to your bid will be based on your compliance with the requirement to list the exceptions applicable to this project only that are necessary due to manufacturing, not cost constraints
- Include the added cost to comply with requirements in your bid and not as an option
- Blanket exceptions to industry specifications (API, ANSI etc.) are not acceptable

Figure 10.18 Important ITB message to all vendors

If the above requirements are strictly enforced, the benefits noted above will result in significant project savings in terms of man hours and schedule time.

Pre-bid activity and degree of audits

Once the ITB has been released to the quoting vendors, the work begins. The first order of business is to prepare for audits required at this stage. If the equipment in question is prototype, design and manufacturing audits have already been initiated and are on going. If the equipment contains multiple major components that do not have field experience, audits are required in this phase. These facts are shown in Figure 10.19 (refer to the design audit section of this chapter for specific details).

Design and/or manufacturing audit requirements in the pre-bid phase

- Finalize audit results and prepare project team recommendations for prototype class equipment
- Interview quoting vendors and determine requirement for design and/or manufacturing audits during pre-bid phase for major equipment with multiple component inexperience

Figure 10.19 Design and/or manufacturing audit requirements in the pre-bid phase

The concept of pre-bidding is very powerful and rewarding to all three parties in the bid process – vendors, contractors and end users. Pre-bidding requires that all technical details are discussed, with appropriate changes for optimum safety and reliability made, before a price is quoted. The advantages of this approach are presented in Figure 10.20.

Technical discussions before $$$$$

Eliminates competitive pressures on the vendor by:

- Allowing technical review before price
- Assures the same scope for each supplier

Assures offering of the highest safety and reliability

Is performed regardless of risk classification

Figure 10.20 Technical discussions before $$$$$

The pre-bid meeting is frequently called a bid clarification meeting. This title can be misleading and may not have the same advantages as a pre-bid meeting. The significant differences are noted in Figure 10.21.

Bid clarification vs. pre-bid meeting differences

- Pre-bid meetings – are conducted before a price is quoted and allow for modifications to technical offering
- Bid clarification meetings – are conducted after a price is quoted and may not allow for modification to technical offering

Figure 10.21 Bid clarification vs. pre-bid meeting differences

It is most important to confirm the requirements and details of the pre-bid meeting with the contractor in the beginning of the project. Figure 10.22 presents the benefits of conducting a true pre-bid meeting and not a bid clarification meeting.

The pre-bid meeting assures

- The highest equipment reliability
- The lowest life cycle cost
- Equal scope of supply for each vendor
- Shortest bid evaluation cycle time

Figure 10.22 The pre-bid meeting

A pre-bid procedure fact summary and a typical agenda for a compressor train are contained at the end of this chapter. It is recommended that this information be used to justify these meetings with the project management team as early as possible in the project, preferably in the pre-FEED phase.

Due to competitive pressures, past union agreements and high in-house manufacturing costs, vendors have been forced to use numerous sub-suppliers for major component and auxiliary system manufacture and in some cases, design. This approach exposes the end user to potential delivery delays due to sub-supplier manufacturing, quality and schedule issues. These important facts are presented in Figures 10.23 and 10.24.

Who really manufactures it?

Vendors frequently use sub-suppliers for:

- Lower component costs
- Reduced vendor machine shop investment
- Greater schedule flexibility
- Reduced in-house shop load

Figure 10.23 Who really manufactures it?

Potential sub-supplier issues

- Component scrap due to inexperience
- Component scrap due to improper machine tools
- Component scrap due to improper handling
- Poor or non-existent inspection
- Delay in shipment

Figure 10.24 Potential sub-supplier issues

Based on the potential sub-supplier problems noted above, when should they be audited? The suggested action is noted in Figure 10.25.

Audit sub-suppliers when:

- Experience for similar components is low
- Equipment risk class is high
- End user 'lessons learned' warrant

Figure 10.25 Audit sub-suppliers

The final recommendation therefore is to always have vendors define major sub-suppliers and their experience during the pre-bid phase. Please refer to Figure 10.26.

Always require definition of sub-vendor and experience for:

- Casing
- Impellers and/or blades
- Diaphragms
- Shaft
- Baseplate
- Auxiliary systems
- Control panels

Figure 10. 26 Always require definition of sub-vendor and experience

At this point, all details concerning vendor experience, scope, exceptions and sub-supplier experience have been identified. If the objectives of the pre-bid phase and any required audits have been met, the bid evaluation phase will be short and easy since there will be a true ‘apples to apples’ comparison and the lowest price vendor can be selected without any additional meetings or discussions.

Bid evaluations

It is an established fact that the contractor will prepare the bid tabulation and present it to the end user for acceptance. However, it has been my experience that the contractor bid tabulation is usually nothing more than a scope of supply list and not a true technical comparison of the offered equipment.

As noted in the last section, if the pre-bid meetings are conducted correctly, the scope and the vendor exceptions to specifications will be essentially the same. Based on my experience, I require the contractor to be sure to include a ‘technical check list’ section in the bid tabulation for a quick confirmation that the best technical alternative, that will produce the highest availability and therefore highest profits, is selected. I have included an example of a technical check list for a compressor train in the back of this chapter.

Based on my experience, I have included a bid tabulation key fact list in Figure 10.27.

Bid tab 'key fact' list

- The end user should review the contractor proposed bid tab format
- A technical check list should be included to quickly identify advantages
- Carefully consider the weight given to power costs and their tendency to influence selection of equipment of lower availability
- Detail exceptions to specifications and require vendors to list only exceptions specific to the project
- Encourage contractor to minimize size of the bid tabulation since scope should be equal based on pre-bid meeting results

Figure 10.27 Bid tab 'key fact' list

Pre-award meeting

After completion of the bid tabulation and approval of the selected vendor, confirmation of the approved vendor's proposal details are required prior to the award of an order. In my experience, there have been many times when the vendor's marketing and engineering departments have had significant differences of opinion in regard to 'what was actually sold'. The purpose of the pre-award meeting therefore is to confirm order content before a contract to eliminate additional costs and delays during the equipment engineering and manufacturing phases. Key facts regarding the pre-award meeting are presented in Figure 10.28.

Pre-award meeting 'key facts'

- Purpose - to assure agreed compliance
- With who? - the recommended vendor
- When? - asap after the bid tab is approved
- Where? - depends on complexity and risk class
- Confirm - marketing to engineering continuity

Figure 10.28 Pre-award meeting 'key facts'

A suggested outline for a pre-award meeting is noted in Figure 10.29.

The pre-award meeting agenda

- Assure the attendance of vendor marketing and project engineer
- Agenda to be prepared by contractor/end user
- Agenda contents:
    - Scope of supply confirmation
    - Clarification and agreement of all exceptions to specifications
    - Resolution of pending design audit issues
    - Confirmation of price and delivery schedule
    - Agreement of minutes and action points

Figure 10.29 The pre-award meeting agenda

The coordination meeting

After the order is placed, the coordination meeting is the first contact between the contractor, end user hopefully and the vendor. This meeting is usually held approximately 4 weeks after the order placement and should be held at the vendor's shop.

In my experience the effectiveness of this meeting is significantly increased if the end user's rotating equipment specialist and/or consultant, senior operator and maintenance engineer are in attendance. Depending on the project management team, it may be necessary to 'campaign' for the attendance of these valuable people. There can be the suspicion that incorporating these individuals will add additional cost to the job. It is my strong opinion that the addition of these individuals will reduce significantly the life cycle cost of the job by incorporating lessons learned and best practices into the job. Refer back to the pre-FEED phase of the job and note that the same individuals were asked to contribute input to the job in this phase.

My 'best practice' is to include a senior operator and maintenance man (millwright) in all phases of the project from pre-FEED up to and including the test phase to assure that all company 'lessons learned' are turned into 'best practices' for the project.

The key facts for the VCM (vendor coordination meeting) are presented in Figure 10.30.

The VCM – key facts

- Purpose – to confirm scope and design
- Design confirmation amount is proportional to the risk class
- If there is any component inexperience, details must be finalized now!
- Location – vendor's shop
- Attendance by: vendor specialists, sub-supplier specialists, contractor specialists, end user specialists
- Timing – approximately 4 weeks after order
- Duration – 2–4 days based on complexity and risk class

Figure 10.30 The VCM – key facts

Vendor coordination meeting key facts are shown in Figure 10.31.

VCM agenda – key facts

- Agenda by vendor approved by contractor/end user
- Agenda to be issued for review 2 weeks before meeting
- Assure that all required design reviews are included
- Inform project team in advance of required attendance
- End user should take detailed minutes
- Review all minutes and acceptance required by all parties, with action point responsibilities and required dates noted prior to adjournment

Figure 10.31 VCM agenda – key facts

A VCM checklist is included in Figure 10.32 for your use to assure that all important facts are covered. Depending upon the risk class of the equipment, this may very well be the last chance for vendor engineering contact prior to the shop test phase.

VCM agenda checklist

- Review and confirm process conditions
- Review and confirm aero, thermo and mechanical design
- Conduct any required design and/or manufacturing audits
- Confirm all major connection locations
- Review machine and auxiliary layouts for maintenance accessibility
- Review preliminary test agenda
- Resolve any outstanding specification issues
- Review vendor and sub-supplier QC procedures (there may be a separate meeting for this activity)

Figure 10.32 VCM agenda checklist

Design and manufacturing audits

Design and manufacturing audits, as previously stated, are required based on the equipment risk class and vendor and sub-supplier design and manufacturing experience level. These audits can be conducted at any phase of the project but the sooner the better. Prototype equipment requires that audits be conducted during the pre-FEED or FEED phase of the project. Today, most projects are defined as MEGA projects since they are designing process units for the largest size ever built and most probably will incorporate single equipment trains that are prototype in nature. Therefore, many projects require that design audits (pre-screening) be conducted immediately upon project start.

Planning and conducting effective supplier design and manufacturing audits require pre-planning and a significant amount of work, but it is certainly worth the effort in terms of increased profits and reduced project schedule. These salient points are noted in Figure 10.33.

Vendor audit requirements

- Detailed agenda, well in advance
- Design audit at vendor's offices with follow-up at end users offices
- Manufacturing audit at vendor's and/or sub-suppliers plants
- End user specialists must participate
- Conduct preliminary end user in-house checks prior to the design audit if possible

Figure 10.33 Vendor audit requirements

When supplier or sub-supplier manufacturing audits are required, suggested action is shown in Figure 10.34.

Manufacturing audit guidelines

- Machining capabilities (max, size capability)
- Balancing capabilities (high speed, max rotor size)
- Size of assembly area
- Shop load status
- Testing capabilities (gas test, full load test, power limits)
- Handling capabilities (max lift, laydown area)
- Shipping capabilities

Figure 10.34 Manufacturing audit guidelines

Suggested design audit activity

1. Risk type	1	2	3	4
2. *Design Checks*				
■ Aero-dynamic	X	X	?	*
■ Thermodynamic	X	X	?	*
■ Rotor response	X	X	?	*
■ Stability analysis (if applicable)	X	X	?	*
■ Seal balance	X	X	?	*
■ Thrust balance	X	X	?	*
■ Bearing loading	X	X	?	*
■ Train lateral analysis	X	X	?	*
■ Torsional analysis (if applicable)	X	X	?	*
■ Transient torsional (if req'd)	X	X	?	*
■ Control system simulations	X	X	X	X
■ System layout – accessibility	X	X	X	X

Key
1 = Prototype
2 = Multiple component inexperience
3 = Single component inexperience
4 = Proven experience for all components

X = Required
? = Optional
*= Not required

Figure 10.35 Suggested design audit activity

In Figure 10.35, I have presented a suggested list of what the design audit should include based on the risk classification.

After conducting the appropriate audits, prompt follow up regarding any action items is required to confirm acceptance of the supplier and/or sub-suppliers and to maintain the project schedule. Figure 10.36 presents these facts.

Design audit summary and follow up action

- Prepare an executive summary of conclusions
- Immediately present to the project team for approval
- Inform vendor's of results
- Prepare vendor follow up meeting agenda

Note action required and follow up as required to maintain project schedule

Figure 10.36 Design audit summary and follow up action

After completion of the required audits, regardless of what project phase in which they are conducted, follow-up document review is essential to confirm that all stated design and manufacturing requirements are met.

Document review

It goes without saying that document review should definitely be timely, within the project schedule and accurate. However, in addition there are other pertinent facts which are presented in Figure 10.37.

Effective document review considerations

- Assure that required review time frames are realistic and then meet them!!!
- Thoroughly review all items
- Question all required items and follow up
- Be especially careful in the final phases of the project to assure that all required vendor changes have been made

Figure 10.37 Effective document review considerations

Testing phase

The testing phase is the last phase in terms of vendor and sub-supplier design and manufacturing involvement in the project and … the last chance to assure the optimum availability of the finished product.

Remember that all of the equipment addressed in this section is most likely custom designed and no matter how much accrued design and manufacturing experience is present, the possibility of some abnormality, hopefully minor, is high. Therefore, it is imperative that this phase be carefully observed and witnessed by the end user team. Figure 10.38 lists important facts surrounding this phase of the project.

The shop test is an opportunity to:

- Confirm vendor proper design and manufacture
- To match field conditions
- To witness assembly and disassembly using job special tools
- To have plant personnel observe test, assembly etc. and take pictures for purposes of emergency field maintenance excellence
- Review the instruction book
- Have the assigned vendor service engineer observe the equipment he will install
- Review all vendor field procedures

Figure 10.38 The shop test

I have included a shop test checklist at the end of this chapter that will be valuable in planning and executing the shop test phase. Yes there certainly are many opportunities to assure equipment reliability during shop test but there are also a lot of potential lost opportunities if they are not justified to the project team early, during the pre-FEED phase, of the project. The potential lost shop test phase opportunities are noted in Figure 10.39.

Potential lost test opportunities

The following opportunities will be lost if they are not justified at project inception:

- Possible full load test
- Unproven component tests
- Attendance at test by plant personnel
- Use of special tools
- Vendor permission for pictures
- Agreement that assigned vendor field service specialists will be present for tests
- Agreement that the instruction book is reviewed
- Agreement for formal field construction meeting to clearly define all vendor procedures from receipt of equipment on site to initial run in of equipment

Figure 10.39 Potential lost test opportunities

The success of the shop test depends on a good test plan that is reviewed by the end user and contractor and modified as requested well in advance of the test. Figure 10.40 presents these facts.

Shop test agenda review – key facts

- The agenda is issued for review 2 months prior to test
- It incorporates agreed VCM scope
- Compressor performance test conditions are per ASME PTC-10 requirements
- A sample of test calculations and report format is included
- Vendor concurs with all end user and contractor comments prior to test

Figure 10.40 Shop test agenda review – key facts

I actually began my career in rotating equipment on the test floor. And I can still remember how we would see the witnesses come in with an intent to completely participate in the entire test only to leave for a long 'test lunch' an hour or so later. Why did this occur? Usually because the concerned end user and contractor witnesses did not have the opportunity to review the test set up and the procedure prior to the test. As a result, I have always been a proponent of a pre-test meeting.

Is it always required? I think it is but the detail and timing of the meeting depends on certain factors. These factors are noted in Figure 10.41.

When is a pre-test meeting required?

- If the equipment is prototype
- If the equipment is complex
- If a full load test is required
- If the test facility is new

Figure 10.41 When is a pre-test meeting required?

If it is decided to conduct a pre-test meeting, the key facts are noted in Figure 10.42.

Pre-test meeting – key facts

- Conduct the meeting prior to the test day
- Send the agenda to the vendor well in advance
- A typical agenda outline:
  - Confirm test agenda requirements
  - Confirm all test parameter acceptance limits
  - Confirm instrument calibration
  - Review test set up or concept drawing
  - Review data reduction methods
  - Confirm all test program agreements

Figure 10.42 Pre-test meeting – key facts

Figures 10.43 to 10.45 define recommended test activity for the mechanical, auxiliary equipment and performance shop tests respectively.

Mechanical test – key facts

- Per API and project requirements
- Confirm all components are installed
- Confirm all accessories are installed
- Monitor progress of test, look for leaks etc.
- Do not accept test until all requirements are met

Figure 10.43 Mechanical test – key facts

Auxiliary system test – key facts

- Must be per API and project requirements
- Confirm that the test agenda is followed
- Confirm all components are installed
- Confirm that all required instruments are installed
- Monitor the progress of the test – look for leaks etc
- Do not accept until all requirements are met

Figure 10.44 Auxiliary system test – key facts

Compressor performance test – key facts

- Per ASME PTC-10 requirements
- Reconfirm test speed is per PTC-10
- Confirm all instruments are calibrated and installed
- Confirm test gas purity
- Agree that conditions are stable prior to each test point
- Confirm vendor's calculations for each test point
- Do not accept until all test requirements are met

Figure 10.45 Compressor performance test – key facts

At the conclusion of all test activities, there is still important work to be performed. These items are defined in Figure 10.46.

Post test – key facts

- Confirm performance results, corrected to field conditions
- Confirm mechanical test acceptance
- Confirm auxiliary system test acceptance
- Inspect components and confirm acceptance
- Agree to any corrective action in writing
- Accept or reject test – any corrective action requires a retest!

Figure 10.46 Post test – key facts

What happens if the test is not successful? Approximately 50% of the tests that I have either run or participated in over my career have not been successful in regards to one component or more not meeting test requirements. Possible rejected test action is noted in Figure 10.47.

Rejected test action

- Immediately provide details to the project team
- Confirm if field conditions can handle the abnormality
- Determine if the 'as tested' machine will meet all reliability requirements
- If the decision is to reject, inform the vendor and detail the reasons
- Do not accept unrealistic delivery delays

Figure 10.47 Rejected test action

Finally, do not forget the importance of test report requirements. The test report is a most important document that represents the 'baseline performance of the unit' and will be a benchmark for field operation acceptability. Test report – key facts are noted in Figure 10.48.

Test report – key facts

- The shop test is the field baseline!
- The test report must be detailed and complete
- Review the preliminary contents of the report before leaving the test floor
- Obtain the actual test results
- When the final report is received, check the results obtained at test against the final report
- Immediately contact the vendor if there are any differences

Figure 10.48 Test report – key facts

Appendix

Suggested vendor pre-bid meeting details letter

The following are suggested letter contents. Comments for consideration are noted in **bold**.

Please be advised that you will be asked to attend a pre-bid meeting at ____________ (Your or Contractor's or Company ... decision required) offices. **Note: If possible, the meeting should be held at the vendor's offices. This is advisable since more experienced specialists are immediately available to answer any questions that may arise. This decision should also be influenced by the machinery risk classification. The higher the risk, the more important the vendor office meeting is.**

The pre-bid meeting will take place approximately _______ **(2–4 weeks after receipt of bid and must be coordinated with the project team ... note this decision will also be influenced by the machinery risk class).** Only technical details will be discussed. Please bring the technical, un-priced proposal for the equipment that you will quote (Trains include compressor, gear (if applicable) turbine and auxiliary systems). Your representatives at the meeting must include an experienced application, instrument engineer and any other personnel you require.

The meeting objective will be to qualify your bid technically based on component experience and to fully define scope of supply and approve exceptions to specifications. If necessary, the technical aspects for your equipment may change as a result of the meeting discussions. In addition to our (contractor) equipment specialist the end user specialist ________ **(or other assigned engineer)** will participate in these meetings. We emphasize that it is in your interest to bring the most qualified personnel to the meeting since this will be the only technical meeting prior to the final bid.

The following additional audits may be required as a result of your bid details and the use of major sub-suppliers.

____________ **(The end user to identify sub-suppliers for**
____________ **manufacturing, handling and shipping audits**
based on vendor bid details)

At the conclusion of the meeting, all details will be summarized and you will be asked to submit your priced proposal in accordance with the technical details, scope of supply and approved exceptions to specifications agreed to in the meeting. Your proposal will be required in _______ (**Normally 2 weeks but may be longer based on complexity of equipment offering and machinery risk classification**) weeks after the meeting.

The pre-bid meeting agenda is attached.

Pre-bid procedure fact summary

The following is a brief summary of the salient procedure facts:

1. **Required personnel experience** – experienced rotating equipment specialists from contractor, supplier(s) and client are required to participate in the pre-bid meetings. Note: end user 'in-house' specialist are required.
2. **Individual supplier meetings** – individual meetings are held with each supplier, using notes from previous meetings, to assure equal supplier experience, scope and exceptions to specifications.
3. **Meeting duration** – Anticipated 1–2 days per major equipment train depending on machinery risk classification. Note: this includes compressors, drivers and auxiliaries. Please refer to the attached typical agenda.
4. **Typical pre-bid meeting activity**
   - Technical details are reviewed using agenda requirements to assure proven component experience (impellers, diffusers, rotor response, bearing, seal, and auxiliary system etc.), scope compliance and acceptable exceptions to specifications.
   - Modifications are made, as necessary, to assure that each vendor is offering proven components within acceptable design limits.
   - Manufacturing capabilities are confirmed and sub-suppliers for all major components and auxiliary systems are identified and their experience is confirmed for similar component manufacture.
   - At the conclusion of the meeting, notes are reviewed and each vendor is instructed to submit a final priced proposal, in full accordance with meeting notes that will be used for the bid evaluation.
   - Depending upon machinery risk classification, additional end user in-house and/or independent 3rd party design checks may be required. In addition, separate vendor and sub-supplier machining, handling and shipping capability audits may be required.

Typical compressor train pre-bid meeting agenda

Please note that the following agenda will be followed for each of the compressor trains being offered. Note: 1–2 days will be required for the meeting to review all details based on unit risk classification.

1. **Compressor experience review (vendor to include necessary reference charts, tables etc.)**
    - Casing experience and review of compressor layout drawing
    - Impeller experience (flow and head coefficient)
    - Individual impeller curve (location of rated point to impeller best efficiency point)
    - Impeller stress
    - Rotor response
    - Stability analysis (if applicable)
    - Bearings – surface speed, load and experience
    - Thrust balance
    - Seals – surface speed, balance forces and experience
    - Surge control and process control system
2. **Steam turbine or motor experience review (vendor to include necessary reference charts, tables, etc.)**
    - Turbine casing experience and review of layout drawing
    - Stage nozzle and blade experience (profile, velocity ratio, BTU/stage)
    - Blade attachment method and blade stresses
    - Campbell and Goodman diagram review
    - Rotor response
    - Bearings – surface speed, load and experience
    - Thrust balance (reaction and hybrid types)
    - Shaft seals
    - Transient torsional response experience review (synchronous motors)
    - Control and protection system

3. **Gear experience (if applicable) (vendor to include necessary reference charts, tables etc.)**
   - Gear box experience review and review of layout drawing
   - Review of gear data sheet
   - Gear calculation review (in accordance with API 613)
   - Bearings – surface speed, load and experience
   - Thrust loading – single helical gears
   - Pitch line velocity review
4. **Auxiliary system experience (lube, dry gas seal and control oil system)**
   - Review of P&IDs
   - Review of API 614 data sheets
   - Review of typical arrangement drawings
   - List of experienced system sub-suppliers
   - Review of proposed dry gas seal supplier information
5. **Scope of supply for compressor train (all components and auxiliaries) review**
6. **Compressor train (all components) exceptions to specification**
7. **Meeting summary and action required.**

   Note: Based on machinery risk, the following 'design checks' may be required:

   - Aero-dynamic
   - Thermodynamic
   - Rotor response
   - Stability analysis
   - Seal balance
   - Thrust balance
   - Bearing loading
   - Control system simulations
   - System layout maintenance accessibility

Typical FAI compressor bid tabulation

Vendor (2)	
Machinery vendor	
Train configuration – layout of machines	
LP compressor: model/type (3)	
HP compressor: model/type (3)	
Turbine: model/type	
Train inlet flow – actual m^3/hr (normal/design) (4)	
Train total power kw (normal/design) (5)	
HP discharge pressure (bar)	
Train cooling water requirements – M^3/hr	
Train (air cooled option) heat rejected – kw	
Train total bearings – radial/thrust	
Number of train lateral and torsional critical speeds	
LP compressor speed Casing type	
LP inlet guide vanes – yes/no	
% IGV for normal/rated case	
Types of compressor elements/number (6)	
Maximum element tip speed – m/sec	
Proven experience for all elements? – yes/no (10)	
Bearing span/diameter under impeller (7)	
Maximum radial bearing specific load – barg (8)	
Rotor response characteristics acceptable? yes/no (9)	
HP compressor Vendor/model	
HP speed	
Casing type	
HP inlet guide vanes – yes/no	
Types of compressor elements/number (6)	
Maximum element tip speed – m/sec	

Proven experience for all elements? - yes/no (10)	
Bearing span/diameter under impeller (7)	
Maximum radial bearing specific load - barg (8)	
Rotor response characteristics acceptable? yes/no (9)	
Total number of gear meshes in the train (11)	
Maximum gear mesh pitch line velocity m/sec	
Gear mesh experience? yes/no (12)	
Steam turbine vendor/model (13)	
Number of blade rows	
Proven experience for all blades and nozzles? yes/no	
Lube oil system flow - liters/minute	
Inlet air filter vendor/type (14)	
All exceptions to specs acceptable? yes/no (15)	
Specification exception details	
Exceptions to component experience? yes/no (10)	
Component experience exception details	
Exceptions to Methanex best practices? yes/no (16)	
Best practice exception details	
Order of technical preference	
Supporting reasons for technical preference order	

Notes:

1. Based on facts from formal bid summary. Objective: To present specific technical facts to determine order of technical preference based on client best practices, FAI experience, component experience and compliance with project requirements (specifications and data sheets).
2. Base, alternate and any vendor process design.
3. Type: Axial, axial-radial, centrifugal, integral gear.
4. Actual flow after the inlet air filter. Normal point is vendor guarantee point. Design point is 110% flow point.
5. With 8.8MW generator load.
6. Axial blades, overhung open or closed impellers.
7. Bearing span divided by diameter of the shaft under the blades/impellers for compressors is a

measure of shaft stiffness that confirms dynamic rotor response acceptability if the ratio is 10 or less. Note: vendor experience will still be the deciding criteria if the value exceeds 10.0.

8. Specific load = load on bearing projected surface area (L x D) in barg. Acceptance criteria are per API 617 or if vendor can demonstrate that predicted values are correlated by test and field operating experience. Ie, Man RIK is acceptable based on proven field and test data.
10. Component experience exceptions are detailed in experience exception cell(s).
11. A gear mesh is defined as the mesh of a pinion and gear regardless if the mesh is in a separate gear box or integral gear compressor casing.
12. Factors considered for gear mesh experience: pinion diameter, tooth width and specific power transmitted.
13. All quoted steam turbines are the hybrid type. Ie, first stage impulse, remaining stages reaction.
14. Type: pulse type (specified), multistage type, rolling element type.
15. Exceptions are detailed in specification exception cell(s).
16. Exceptions are detailed in best practice exception cell(s).

Shop test checklist

1. Scope
   - Appropriate industry specs included (ANSI, API, NEMA etc.)
   - In-house and/or E&C specs included
   - Project specific requirements
     - ☐ Performance test ☐ All rotors ☐ One rotor
     - ☐ Test (equivalent) conditions
     - ☐ Field (actual) conditions
     - ☐ Mechanical test ☐ All rotors ☐ One rotor
     - ☐ Test (equivalent) conditions
     - ☐ Field (actual) conditions
     - ☐ Unit test of all equipment (string test)
       - ☐ No load ☐ Includes auxiliary systems
       - ☐ Full load ☐ Does not include auxiliary systems
     - ☐ Use of job couplings and coupling guards
     - ☐ Testing of instrumentation, control and protection devices
     - ☐ Auxiliary system test
       - ☐ Lure oil ☐ Test press ☐ Full press
       - ☐ Control oil ☐ Test press ☐ Full press
       - ☐ Seal oil ☐ Test press ☐ Full press
       - ☐ Seal gas ☐ Test press ☐ Full press
       - ☐ Fuel ☐ Test press ☐ Full press
       - ☐ Flow measurement required
       - ☐ Time base recording of transient events required
     - ☐ Use of all special tools during test (rotor, removal, coupling, etc.)
     - ☐ Shop test attendance (includes assembly and disassembly)
       - ☐ Site reliability ☐ Site maintenance ☐ Site operations
       - ☐ Review of instruction book during shop test visit
       - ☐ Test agenda requirements
       - ☐ Mutually agreed limits for each measured parameter
       - ☐ Issue for approval 2 months prior to contract test date

2. Pre-test meeting agenda
   - Meet with test department prior to test to:
     - ☐ Confirm test agenda requirements
     - ☐ Confirm all test parameters have mutually agreed established limits
     - ☐ Review all instrument calibration procedures
     - ☐ Review test set-up drawing
     - ☐ Review data calculation (data reduction) methods
     - ☐ Define work scopes for site personnel (assembly and disassembly witness, video or still frame pictures, etc.)

- ☐ Confirm assigned vendor service engineers will be in attendance for:
  - ☐ Assembly
  - ☐ Disassembly
  - ☐ Test

Shop test activity

- ■ Review and understand test agenda prior to test
- ■ Immediately prior to test meet with assigned test engineer to:
  - ☐ Review schedule of events
  - ☐ Designate a team leader
  - ☐ Confirm test team leader will be notified prior to each event
  - ☐ 'Walk' test set-up to identify each instrumented point
  - ☐ Confirm calibration of each test instrument
  - ☐ Obtain documents for data reduction check – if applicable (flow meter equations, gas data, etc.)

- ■ During test

  Note: coordinate with test personnel to avoid interference
  - ☐ Review 'as measured' raw data for consistency
  - ☐ 'Walk' equipment – look for leaks, contract instrument, piping and baseplate vibration, etc.
  - ☐ Use test team effectively – assign a station to each individual
  - ☐ Ask all questions now, not later, while an opportunity exists to correct the problem
  - ☐ Check vendor's data reduction for rated point – if applicable

- ■ After test
  - ☐ Inspect all components as required by the test agenda (bearings, seals, labyrinths, RTD wires, etc.)
  - ☐ Review data reduction of performance data corrected to guarantee conditions
  - ☐ Review – all mechanical test data
  - ☐ Generate list of action (if applicable) prior to acceptance of test
  - ☐ Approve or reject

11

Machinery installation guidelines

- Introduction
- Site procedure
- Foundations
- Piping
- Shaft alignment
- Couplings
- Cleaning of equipment and associated pipe
- Final inspection and start-up checks
- First start, run in and initial operation

Appendix

- Auxiliary system flushing procedure
- Best practices for developing an effective site pre-commissioning program for rotating equipment
- Functional lube/seal system test procedure outline
- Electro-hydraulic governor functional test procedure outline
- Steam turbine solo run functional test procedure outline

Introduction

Regardless of the quality of design and manufacture, regardless of a successful test and efficient shipment, installation will determine the amount of maintenance required and the resulting revenue of the process unit. Figure 11.1 shows the general site considerations that are

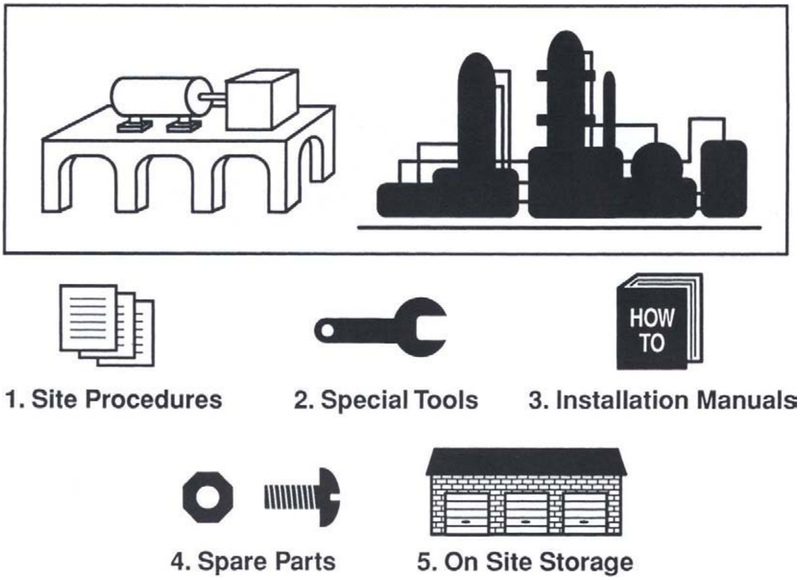


Figure 11.1 General site considerations

required for a successful field installation. Each one of these items will now be covered.

Site procedures

The importance of site installation procedures cannot be overemphasized. Figure 11.2 shows the most commonly required site installation procedures.

It must be remembered that the objectives of the construction contractor and of the end user are identical in terms of profit. However, they are dissimilar in means to achieve their common objectives. The contractor's objective is to construct a safe and reliable process unit within the budget and on time. This objective is opposed to the end user's objective, who must operate the process unit for thirty years or more at maximum profit and thus requires maximum reliability of the installed equipment. The only leverage that the end user has in meeting his objectives is to require practical, proven site installation procedures that will result in the most reliable, safe and cost effective installation of his equipment. Frequently the contactor will rely on the equipment vendor to provide most of the site procedures. It is strongly recommended that the end user, early in the project, require approved procedures for every major site installation milestone. These procedures include, but are not limited to:

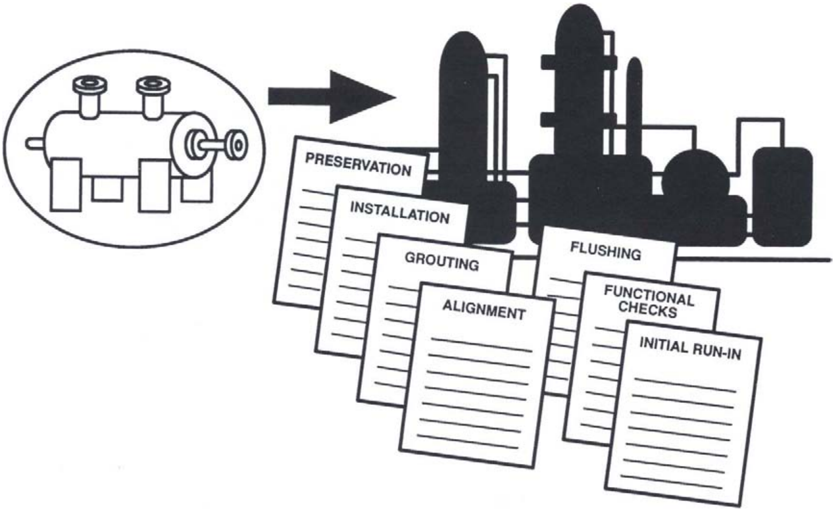


Figure 11.2 Site installation procedures

- Equipment preservation
- Equipment installation
- Grouting
- Alignment
- Flushing
- Functional checks
- Initial run in of equipment

It must be mentioned that preservation procedures are often ignored early in a construction project and become written and implemented too late to effectively prevent equipment deterioration due to corrosion. Again, require procedures be written and approved well in advance in the start of construction.

A specific 'initial run in procedure' for each major piece of rotating equipment should be reviewed by the end user well in advance. It is recommended that the end user review these procedures during the shop test with vendor field service engineers to assure that the Run In procedure is in accordance with vendor and plant best practices. An example of not performing a pre-Run In review and its consequences, concerned a large high pressure condensing steam turbine cold start-up time vs. speed curve. Since the turbine was designed for an automated sequenced start, the cold start-up curve was programmed into the PLC without detailed review by the end user. The curve was simulated and did not take the specific parameters of the application into account. The result was a severe rub that damaged rotor and stationary internals and resulted in over a 40 day plant start-up delay. The daily profit of this plant was approximately 0.25 $MM.

Construction special tools

Most of the equipment that is installed will be custom designed. Therefore, special tools will be shipped with equipment that can only be used for that particular item. Consequently, these tools must be listed and stored in a proper location that will allow maintenance personnel to easily locate these tools when required. Many of the tools such as hydraulic jacks, special mounting devices, etc. also will require preservation during storage to prevent corrosion. Many times this requirement is overlooked. **Be aware!**

An example of not properly storing special tools and spare parts is a high humidity, tropical island installation where the spare rotor and coupling hydraulic mounting tools were to be used for a turnaround. They were not inspected prior to the turnaround and the result was that the spare rotor could not be installed due to excessive corrosion. The coupling hydraulic mounting adapter had to be replaced with a new adapter due to excessive corrosion. It should be noted that this equipment was stored in a sealed container with a nitrogen purge but unfortunately the seal was faulty and the nitrogen purge pressure was not monitored.

Installation manuals

Like site procedures, installation manuals frequently arrive after they are first required. The installation manual will contain valuable information pertaining to receipt of equipment, preservation, interim storage and of course, installation. It is recommended that the end user again require that manuals be approved and received well in advance of the start of any installation activity.

In an effort to assure that the instruction manual contains information that is accurate and specific to the job, we have written into the job specification that the instruction book shall be completely reviewed at the time of the shop test and shipped with the unit. Almost every such review has uncovered incorrect information and/or general information that is not specific to the particular project that would have resulted in confusion during disassembly/assembly of the equipment leading to possible equipment reliability problems.

Spare parts

The end user must require the contractor to be responsible for the proper storage of spare parts on site and most importantly, the receipt of all required start-up spares, operating spares and capital spare parts well in advance of the start of construction activity. Many times spare parts are required prior to initial operation of equipment since components are broken during shipment.

It has been our experience that many of the spare rotors for major, un-spared equipment are not properly inspected and maintained upon receipt from the vendor. A specific rotor container inspection procedure should be written by the contractor and approved by the vendor and end user to assure that all rotor preservation will be maintained from initial receipt date on site. There have been many cases where the rotor containers have never been inspected prior to the intended use date and could not be used and had to be returned to the vendor for rework.

On site storage

Prior to the arrival of any equipment on site the contractor should review the manufacturer's requirements and provide extended on site storage facilities that meet or exceed those requirements. In addition, the contractor shall order and have available the required preservation compounds.

When selecting the preservative compounds, special care should be given to the selection of compounds for the specific site environment and should be based on local experience. In humid, seacoast and offshore environments, special care should be given to assure the preservative compounds can resist high moisture/salt environments. In addition, components should be checked frequently to confirm that the compounds are providing the required protection.

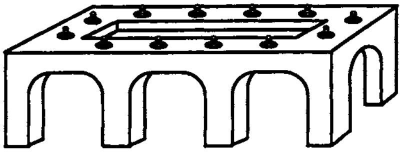
Foundations

The installation of properly designed and constructed foundations play an important part in the long term availability of equipment. This section will cover the major aspects of sound foundations.

General considerations

After the proper civil work is done, and the foundation is designed in accordance with specifications, there are certain general considerations required. These are shown in Figure 11.3.

Foundations



General considerations

- Rough surfaces
- Sufficient space for grout
- Sole plates

Figure 11.3 Foundations

Foundations must be rough enough to allow grout to adhere. The elevation of the top surface should allow at least one inch of grout under the base plates or sole plates. When machinery is mounted directly on the foundation, sole plates must be provided. It is wise to epoxy grout sole plates to facilitate easy removal and installation of equipment during maintenance. Sole plates must be level in themselves and all other planes.

Foundation bolts

Each equipment manufacturer has foundation bolt requirements. It is a good idea to review the contractor's foundation bolt arrangements prior to the start of any equipment installation and assure that the contractor's procedure, types of bolts and bolt arrangement, meet or exceed the equipment vendor's requirements. Many a project has been delayed by not incorporating this requirement. Figure 11.4 shows three typical installation arrangements for anchor bolt installations.

A case history for the installation of a large reciprocating compressor in a refinery shows how poor planning can cause a significant construction delay. The foundation re-bar pattern was not coordinated with the foundation bolt pattern for the crankcase and crossheads. After the foundation was set, with the foundation bolts in place, it was discovered that the bolt locations had moved from the original positions and that the crankcase could not be positioned over the foundation bolts. The re-bar had interfered with the foundation bolts causing them not to be correctly positioned. The result was complete foundation rework to correctly position the foundation bolts that resulted in a delay of one month to the construction schedule. At that time the lost revenue was approximately 1$MM per day.

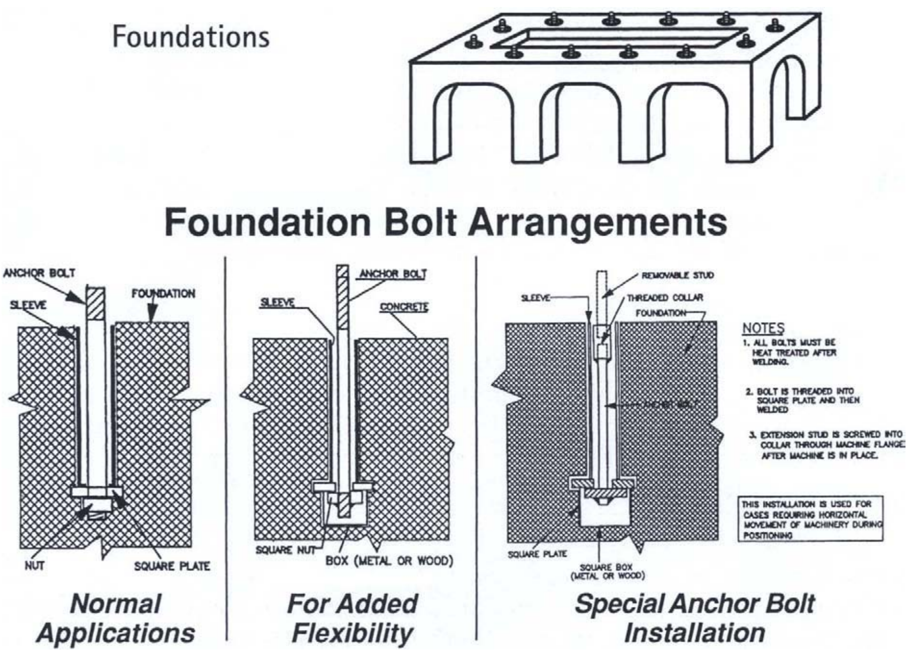
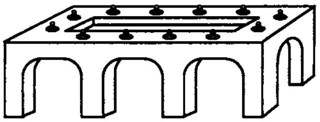

Foundations
Foundation Bolt Arrangements
ANCHOR BOLT
SLEEVE
FOUNDATION
NUT
SQUARE PLATE
Normal Applications
ANCHOR BOLT
SLEEVE
CONCRETE
SQUARE NUT
BOX (METAL OR WOOD)
For Added Flexibility
REMOVABLE STUD
SLEEVE
THREADED COLLAR
FOUNDATION
ANCHOR BOLT
SQUARE PLATE
SQUARE BOX (METAL OR WOOD)
NOTES
1. ALL BOLTS MUST BE HEAT TREATED AFTER WELDING.
2. BOLT IS THREADED INTO SQUARE PLATE AND THEN WELDED
3. EXTENSION STUD IS SCREWED INTO COLLAR THROUGH MACHINE FLANGE AFTER MACHINE IS IN PLACE.
THIS INSTALLATION IS USED FOR CASES REQUIRING HORIZONTAL MOVEMENT OF MACHINERY DURING POSITIONING
Special Anchor Bolt Installation

Figure 11.4 Anchor bolt installations (Courtesy of ME Crane Consultant)

One last word regarding foundation bolts. It is easy to mis-locate the bolt locations relative to the machinery base plate holes. Before bolt holes are randomly elongated to facilitate misplaced location bolts, all facts should be discussed with both the contractor and the vendor of the equipment. Irresponsible action regarding elongation of bolt holes have caused machinery problems. Fabricated foundation bolts on which welding is used in the fabricated assembly must be stress relieved after welding.

Leveling

Figure 11.5 presents the basics of leveling of equipment.


Foundations

1. Leveling of equipment within a tolerance of:
   - ±0.0025 in/ft (0.05 mm/m)
   - Confirm with a calibrated engineer's level (1 division = 0.001 in/ft)
2. Back off all jackbolts and chocks minimum of 2 turns

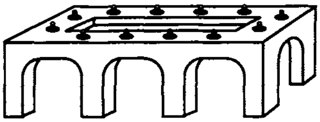
Figure 11.5 The basics of leveling of equipment

Equipment must be leveled within a tolerance of .05 mm per meter and confirmed with a calibrated engineer's level. Any special leveling instructions given by the vendor must be followed. In the case of reciprocating equipment, it is important that shims straddle hold down bolts. When jacking screws are used for leveling equipment it is not necessary to remove them after grouting but they must be backed off at least two turns and the hold down bolts must be retorqued to their correct value after grout has adequately cured. There should be a minimum of one jacking bolt for each hold down bolt.

Grout (general)

The grouting plays an important role in the availability of equipment. Improper grout type and application of the grout has caused many an unscheduled shutdown in the field. Figure 11.6 presents some general grout considerations that have been proven through many long, hard construction projects.

Foundations



General grout considerations

1 Approved grouting procedure

2 Epoxy grout for:
  - Greater than 100 hp (750kw)
  - All axial, centrifugal or reciprocating compressor trains

3 Special environmental conditions
  - Temperatures >140°F (50°C)

4 Proper surface preparation
  - Clean
  - Chipped
  - Water free (for epoxy grout)
  - Grease anchor bolts, jackbolts, chocks

Figure 11.6 General grout considerations

Most important is an approved grouting procedure. Not a simple procedure that states the type of grout and how much will be used, but a detailed procedure specifying the equipment used for proper grout pours, the forms, the form preparation, the details concerning depth of pour, specifications for grout, etc. It has been our experience that contractors are not experienced in proper grouting procedures. Remember, the installation phase is only a short period in the life of equipment. The decisions made during grouting will affect the equipment for its lifetime.

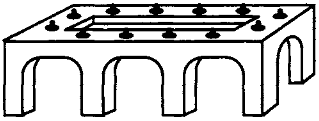
Epoxy grout is usually required for equipment greater than 100 horsepower and all reciprocating types of rotating equipment. Although epoxy grout is much more expensive than conventional grouts, it certainly pays out in the long run since it is impervious to oil and resists cracking. In the application of any grouts, ambient conditions are very important. Be sure that the site grouting procedure takes the local ambient conditions into account.

Like most jobs, proper preparation significantly affects the quality of the finished product. Clean, chipped, water free (for epoxy grout) foundations are a necessity. Also anchor bolts, jack bolts and chocks should be greased for ease of operation once the grout starts to cure.

Epoxy grout

Epoxy grout is clearly the grout of choice for critical (unspared) equipment installation since it lasts the longest and is impervious to most external sources. Figure 11.7 presents some epoxy grout considerations.

Foundations



Epoxy grout considerations

1 In accordance with grout manufacturers

2 Wax or grease all forms

3 Limit thickness of pour to 4"

4 Fill bolt sleeves or pockets

5 Check for voids. Fill with epoxy pressure grout

6 Seal grout holes

Figure 11.7 Epoxy grout considerations

It is important that epoxy grout be poured in accordance with the grout manufacturer's recommendations. Most contractors need experience in epoxy grout installation. In fact, it has been our experience that an on site demonstration of epoxy grout by the epoxy grout manufacturer is a worth while expenditure and saves countless repours and project delays. Some other epoxy grout considerations are:

- Wax or grease all forms
- Limit thickness of pour to approximately four (4) inches
- Assure that proper mixing and pouring tools are available
- Fill bolt sleeves or pockets to assure that grout does not spill into these areas
- Check for voids and fill with epoxy pressure grout when required
- Seal grout holes in metal base plates

Non shrink or cementous grout

Frequently for cost considerations, contractors will attempt to use non shrink or cementous grout on large pieces of equipment. In certain instances, this is acceptable as in the case of large oil console foundations which are usually installed with cementous grout for reasons of mass. This action solidifies the console base and significantly minimizes pipe and component vibration.

It should be mentioned that some types of non shrink grout incorporate metal filings. It has been found that in some instances these filings will corrode with time and cause separation of grout from the foundation. Prior to application of any grout, a proper procedure and details of the grout must be defined. In the event of any doubts ask the original equipment manufacturer regarding his considerations.

Piping

Having just discussed foundations which significantly affect machinery availability since the machinery is connected to the foundation, we will now discuss piping. Piping accounts for the other connection of the machinery to the environment surrounding the equipment. Improper piping assembly, like improper foundation installation has resulted in reduced rotating equipment reliability.

General practices

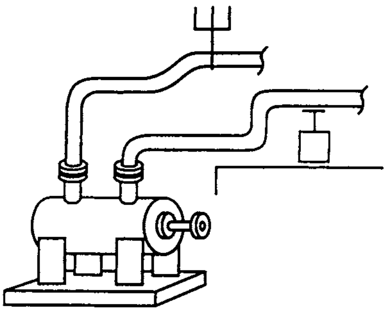
Figure 11.8 presents piping considerations that will result in proper installation of equipment of high reliability.

Piping considerations

1 *Float pipe* to machine. Machine *is not* a pipe support

2 Lock pins on spring or pipe supports adjacent to machinery kept in place until system filled

3 Flanges parallel on machined surfaces and no come alongs!



±0.010 Across face diameter

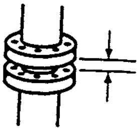


Figure 11.8 Piping considerations

Piping must always be floated to the machine and NOT first mounted on the machine. A machine is definitely not a pipe support. During construction, observe that piping is first mounted to vessels in pipe racks and then and only then floated to the machine. Any other procedure is totally unacceptable.

In bolting piping to machinery, lock pins on springs or pipe supports must be kept in place until the system is filled with liquid or gas since they are designed to support the piping during the operation of equipment. Installation of piping to equipment flanges must be performed with care. Bolts should be freely removed from mating piping and equipment flanges without the use of force (come alongs). Most importantly, flanges must be parallel on the machine surfaces of the mating and equipment flange within +/– 0.010 across the face diameter. It is wise to always observe if piping has been removed or reassembled during turnarounds. Frequently when this activity is performed, proper procedures are not followed and excessive stresses are exerted on the equipment casing. Since the equipment casing supports the bearings which ultimately support the shaft by anti-friction bearings or a thin oil film in the case of hydrodynamic bearings, improper piping assembly can significantly affect machinery operation. Keep this fact in mind.

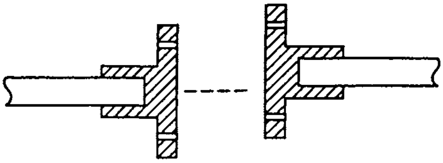
Shaft alignment

It is not the intention of this section to present shaft alignment procedures in detail, but to present considerations that will result in reliable and safe equipment operation.

General considerations

Some general considerations are presented in Figure 11.9 concerning shaft alignment.

Shaft alignment – Preliminary considerations



1 Obtain thermal shafts and machine growth calculations establish 'cold offsets'

2 Coupling hubs installed in accordance with OEM's procedures

3 Set proper B.S.E. dimension

4 Test for 'soft foot' 0.002 maximum allowable differential rise

5 Use only stainless shims – minimize number of shims

6 Shims must straddle hold down bolts

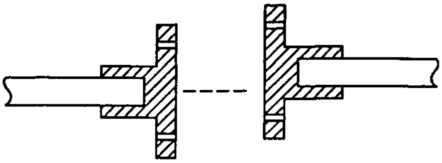
Figure 11.9 Shaft alignment – Preliminary considerations

It is important that proper dimensions be obtained from the equipment manufacturer's instruction book regarding cold offsets and axial separation distance between shaft ends. Many times equipment is presented to the user by the contractor for acceptance when shaft end dimensions are not in accordance with manufacturer's requirements. Proper attention early in the construction phase will alleviate this situation. Coupling hubs must also be installed in accordance with the vendor's procedures. Prior to beginning alignment checks the base plate must be checked for soft foot. This is accomplished by alternately loosening hold down bolts and checking that a maximum differential rise of .002 of an inch is not exceeded. Stainless steel shims should be used only and the number minimized since shims act like a spring. It is the intention to rigidly support equipment. Shims also must straddle hold down bolts to provide maximum surface area and thus minimize high localized stresses on grout and foundation.

Alignment parameters

Regardless of the type of alignment procedure used the alignment parameters presented in Figure 11.10 should be followed as a minimum.

Shaft alignment – Alignment parameters



1 All piping must be disconnected

2 Reverse dial indicator method shall be used

3 The maximum allowable alignment limits shall be 0.002 T.I.R. per inch of spacer parallel offset

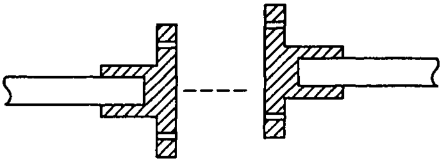
4 Demonstrate results are repeatable

5 Maximum bracket sag = 0.003

Figure 11.10 Shaft alignment – Alignment parameters

Piping must always be disconnected for initial alignment. Results should be repeatable and any alignment brackets should be constructed such that their sag is minimized. As previously mentioned, external piping can cause significant stresses in the casing and eventually the bearing brackets, thus affecting equipment internal alignment. It is imperative that alignment change limits be instituted when connecting piping. The maximum allowable movement of the equipment shaft when piping is connected should be limited to .001". Figure 11.11 presents other details concerning the proper installation of piping to equipment.

Shaft alignment – alignment change limits when connecting piping



1 Mount dial indicators independent of machinery horizontally and vertically reading on coupling flanges

2 Zero out indicators

3 Tighten using specified sequence and torque values

4 Maximum shaft movement = 0.001

5 Dowell if required by OEM

Figure 11.11 Shaft alignment – alignment change limits when connecting piping

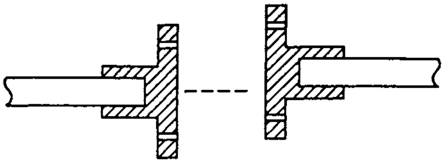
Dowelling

Both driver and driven equipment should be dowelled in the field when specified by the manufacturer. No dowelling is to be done until all piping and alignment is finalized. Dowels with threads on the top end are preferred to facilitate removal without damage to dowels or machinery.

Couplings

The function of a coupling is to transmit torque from the driver to the driven equipment and to allow for relative movement of the connecting equipment shafts. Figure 11.12 presents some important coupling considerations.

Coupling considerations



1 Inspect when received

2 Store special mounting tools (pump, adapters, H.P. tubing etc.)

3 Shall be installed in strict accordance with manufacturers procedure

4 Record the following values:
- B.S.E. distance
- Hydraulic pressure applied
- Axial drive (push-up)
- Final position
- Amount of prestretch (flexible coupling)
- Axial float and position of spray nozzles (gear couplings)

5 Apply proper bolt torque

Figure 11.12 Coupling considerations

The coupling is a part of a system that transmits torque from the driver to the driven piece of equipment. Therefore the proper installation of the coupling is of utmost importance. Important in coupling installation are the special tools (hydraulic pump, adapters, high pressure tubing, etc.). These tools must be stored in an acceptable area and be available when required. The coupling must be installed in strict accordance with the manufacturer's procedure. Of particular importance today are dry couplings (those couplings that do not require lubrication). In order to facilitate flexibility, these couplings employ highly stressed flexible elements. As a result, close attention to proper distance between shafts, coupling installation and location is extremely important and will significantly affect the life and reliability of the couplings that are used.

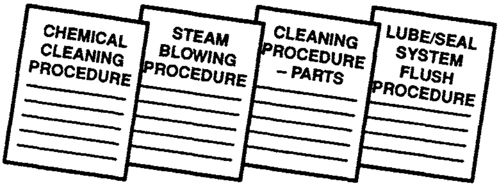
Cleaning of equipment and associated pipe

Many machinery wrecks have been caused by debris originating in associated equipment piping. Thorough cleaning and inspection of piping prior to installation on equipment is a necessity. This section will cover details concerning cleaning various types of associated machinery piping.

Chemical cleaning

Figure 11.13 presents some considerations concerning chemical cleaning of piping.

Cleaning of equipment and associated piping


CHEMICAL CLEANING PROCEDURE
STEAM BLOWING PROCEDURE
CLEANING PROCEDURE – PARTS
LUBE/SEAL SYSTEM FLUSH PROCEDURE

Chemical cleaning of piping

1 Approved contractor cleaning procedure
  - Components to be cleaned
  - Cleaning methods and sequences
  - Equipment required
  - Location of vents and drains
  - Passivation materials and procedures
  - Dry out and blow down

2 Prepare procedure in advance

3 All equipment, personnel, engineering and supervision by contractor

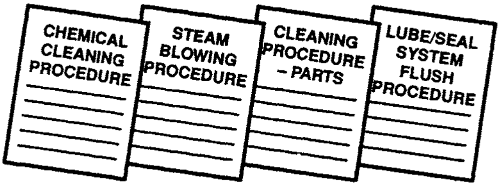
Figure 11.13 Cleaning of equipment and associated piping – chemical cleaning of piping

It is important to prepare the cleaning procedure well in advance and approve that procedure prior to the initiation of any work. As a minimum the procedure should include the details noted. The contractor should be required to provide all equipment personnel and supervision required to perform the cleaning procedure.

Steam piping

Significant damage has been caused to steam turbines by improper steam piping cleaning. It must be remembered that large amounts of debris can be contained in piping flanges and will not be released until the piping system has been thermally cycled. Therefore proper steam pipe cleaning requires a procedure that will expose the piping to all of the stresses and forces that it will encounter during normal long term operation. Figure 11.14 presents some specifics concerning steam blowing of piping.

Cleaning of equipment and associated piping


CHEMICAL CLEANING PROCEDURE
STEAM BLOWING PROCEDURE
CLEANING PROCEDURE - PARTS
LUBE/SEAL SYSTEM FLUSH PROCEDURE

Steam blowing of piping

1 Sturdy target brackets

2 Polished 304 or 316 stainless steel targets

3 Sequence as follows:
- Disconnect upstream of TTV
- Remove strainer
- Install directional stub
- Install targets
- Blow 15 minutes (80% of design, P,T, flow)
- Blow minimum of 10 times
- Allow to cool after each blow
- Repeat until targets are clean (less than (3) particle impressions per centimeter)

Figure 11.14 Cleaning of equipment and associated piping – Steam blowing of piping

There is a common argument between contractor and user that long term steam blows are not required. Be aware that the user and not the contractor will be affected if debris enters the turbine and causes unnecessary shutdowns resulting in lost product revenue.

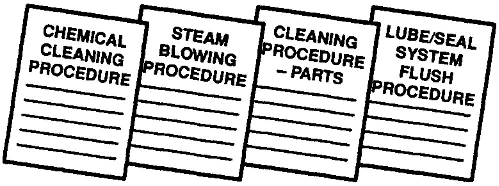
A case history regarding improper steam blowing was the extensive damage rendered to a high pressure condensing steam turbine during initial site operation due to weld slag lodged in the steam drain orifice between the inner and outer casing in the high pressure section of the turbine. This debris caused the retention of condensate in this section of the turbine that did not allow the lower half of the inner casing to expand at the required rate and resulted in an outage of over 6 weeks to repair the inner case and install the spare rotor.

Cleaning of equipment

There comes a time in the installation of all equipment that individual component parts must be cleaned (bearings, labyrinths, seals, etc.). Be aware that cleaning must be performed under controlled conditions so that foreign matter is not introduced.

Figure 11.15 presents considerations concerning cleaning of equipment and related parts.

Cleaning of equipment and associated piping


CHEMICAL CLEANING PROCEDURE
STEAM BLOWING PROCEDURE
CLEANING PROCEDURE – PARTS
LUBE/SEAL SYSTEM FLUSH PROCEDURE

Cleaning of equipment and related parts

1 Use suitable non-flammable safety solvent

2 Paint thinner, gasoline, similar liquids not permitted

3 Preserve parts after cleaning

Figure 11.15 Cleaning of equipment and associated piping – cleaning of equipment and related parts

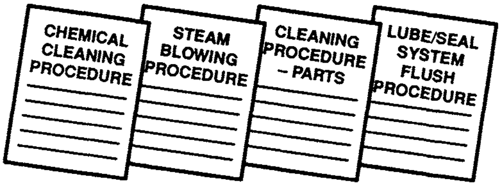
A case history regarding improper inspection after cleaning is the high bearing temperature experienced in a backpressure steam turbine during the initial site solo (uncoupled) run. High bearing pad temperature (95°C) and shaft vibration (25 microns) was experienced in the drive end bearing and was compared to the value experienced during the shop test (70°C) and (10 microns). The vendor representative attributed the increase in temperature to the possible unbalance of the coupling moment simulator (used to simulate the half weight of the coupling spacer). However, when the same situation appeared during the coupled compressor run, I required the bearing to be inspected. It was discovered that the bearing supply orifice was plugged with debris. The bearing was cleaned, inspected for proper clearance and damage (no abnormalities found) and the unit rerun. All results matched the original shop test results and the unit was accepted.

Flushing of lube oil and seal oil systems

As previously stated, all critical equipment incorporates bearings that continuously support the shaft on an oil film approximately 15–20 microns thick. Even though the lubricating and lube oil systems incorporate many components responsible for supply of clean, cool oil at proper pressure, temperature and flow conditions, fine debris existing in pipes between flanges and gaskets, in voids of coolers and other vessels can supply fine metallic and non-metallic particles that can cause significant bearing damage and damage to the equipment. It is therefore imperative that a cost effective flushing procedure be implemented in the field. We have included a cost effective proven oil system flushing procedure at the end of this chapter that has proven time and time again to save valuable construction time and result in producing the cleanest possible oil system that will minimize filter changes during operation.

Figure 11.16 provides some considerations regarding proper flushing procedures.

Cleaning of equipment and associated piping


CHEMICAL CLEANING PROCEDURE
STEAM BLOWING PROCEDURE
CLEANING PROCEDURE – PARTS
LUBE/SEAL SYSTEM FLUSH PROCEDURE

Flushing of lube and seal oil systems

Procedure outline (refer to Appendix)

1 Hand clean major components (lint free cloths)

2 Maintain flushing log

3 Use auxiliary system components

4 Add temporary jumpers and hand valves

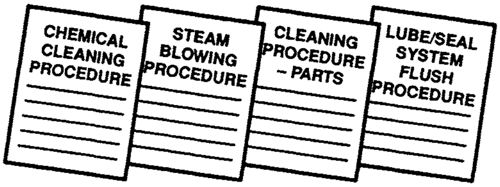
5 Minimize screens (suggested locations)
  - Before and after cooler (1st flush)
  - Supply lines (at jumpers)
  - Return (initial flushes only)

Figure 11.16 Cleaning of equipment and associated piping – flushing of lube and seal oil systems

Again, it must be pointed out that the objectives of the contractor and the end user are different. Maintaining a flushing log is important to assure that all flushing procedure requirements are carried out. In addition, periodic inspection of flushing screens is suggested to assure the job is proceeding smoothly. It is a common occurrence that the length of a flushing cycle frequently exceeds its predicted time. The procedure provided in the appendix and the guidelines presented in this section will assure that flushing will be accomplished in the minimum amount of time. However effective a procedure is, it is only as effective as its implementation. Therefore, monitoring the flushing operation is imperative.

Figure 11.17 presents additional considerations regarding flushing.

Cleaning of equipment and associated piping


CHEMICAL CLEANING PROCEDURE
STEAM BLOWING PROCEDURE
CLEANING PROCEDURE - PARTS
LUBE/SEAL SYSTEM FLUSH PROCEDURE

Flushing of lube and seal oil systems (cont)

Procedure outline (refer to Appendix)

6 Screens (100 mesh SS, 60 mesh backup)

7 Heat cycle

8 Nitrogen bubble

9 Final flush
- white cloth
- acceptance criteria
  - non metallic
  - less than 74 micron

10 Remove all screens, visually inspect, replace all components, install new cartridges, fill reservoir

Figure 11.17 Cleaning of equipment and associated piping – flushing of lube and seal oil systems (cont)

One final word regarding flushing. It has come to our attention that frequently in an effort to assure totally clean systems, too many flushing screens are employed, resulting in excessive flushing times. A common example of this practice is the installation of flushing screens in drain lines. Since drain lines are designed to operate only half full, a flushing screen will eventually become partially plugged. When it does, the level of the oil in the affected drain line will rise and will very effectively clean the top of the pipe. Since this is an abnormal occurrence and since one function of the reservoir is to contain sludge and prevent it from re-entering the system, excessive use of screens in drain lines should be discouraged.

A practical recommendation would be to install one screen in the main return to the reservoir for only the initial phases of flushing. A convenient way to assure that this screen is not becoming plugged is to connect a temporary piece of plastic tubing to the bottom of the drain line close to the flushing screen and monitor the level of the oil. When the level begins to rise in the plastic tube, shut down and clean the screen before the level becomes excessive. Once the debris on the screen has leveled out, it is suggested that the screen be removed.

Final inspection and start-up checks

The contractor and the customer must mutually agree on an acceptable hand over and sign off procedure. As a minimum the following items should be completed.

- All installation and long term storage requirements
- All auxiliary piping, installation, insulation and painting
- All instrument and electrical loop checks complete and proven operable
- All spare parts received and equipment documentation available

First start, run in and initial operation

After the equipment has been handed over, first start, run in and initial operation should be performed in accordance with proven start-up procedures and planned operation procedures. Figure 11.18 provides these considerations.

First start, run-in and initial operation

1. Final inspection and hand-over accepted (all work complete, all punch list items complete, loop checks complete)
2. Start-up procedure (optional) approved
3. Air run (compressors) optional
   - Procedure
   - Safety considerations
   - Limits of operations

Figure 11.18 First start, run-in and initial operation

Included at the end of this chapter are the following additional documents that contain valuable guidelines for site pre-commissioning and commissioning:

- Site pre-commissioning best practice list for your use
- Functional lube/seal system test procedure outline
- Electro-hydraulic governor functional test procedure outline
- Steam turbine solo run functional test procedure outline

Appendix

Auxiliary system flushing procedure

Courtesy of M.E. Crane Consultant

The following procedure is presented as a guide for field flushing of lube and seal systems. In order to be fully productive, it is recommended that all requirements noted herein be strictly followed.

1. General
   - 1.1 Flushing operation will be carried out by the designated party. (Contractor or Owner)
   - 1.2 Cleanliness of oil console, equipment skid, overhead seal oil tanks, piping systems and screens shall be determined by mutual agreement between equipment vendor, contractor and owner.
   - 1.3 Owner and vendor shall keep a log for general review of flushing progress. Master flow sheets shall be kept by owner and updated to progress. An entry shall be made during each shift.
   - 1.4 In general the oil flush shall be performed using selected permanent auxiliary equipment which is part of the vendor supply package. This will include the following:
     - Auxiliary oil pump (electrical) and main oil pump if possible.
     - Main oil filter to be in position for all flushing.
     - Main oil reservoir, degassing tank, overhead seal oil tanks and seal oil traps.
     - Skid piping.
     - Selected instruments and controls.

2 Preparation
   - 2.1 Any residual oil from factory testing must be removed from the reservoir and filters. Relief valves must have been checked prior to flushing.
   - 2.2 The reservoirs, degassing tanks and filter casing must be wiped clean, inspected and approved by the owner. All cleaning must be carried out using a lint free cloth. When filters are open for cleaning, special care should be taken to avoid contaminates falling into 'clean' side of filter housing.
   - 2.3 The filters must be verified to be in place and satisfactory for flushing. Examination of filters will include checks for bypassing, inside or outside of filter housing.
   - 2.4 All lube and seal oil interconnecting piping will be installed consistent with normal operating conditions in accordance with bypass piping arrangements as mutually agreed upon between vendor and owner.
     - All equipment supply and drain piping is required to be flushed during entire flushing operation. Location of all valves, bypasses and screens shall be in accordance with marked-up P&ID's of lube oil, seal oil and control oil system.

- Add hand valves suitable to meet the operating requirements during flushing to *all* piping supply points.
- All lines to the steam turbine throttle valve, servo motors and dump devices will be flushed in accordance with a manufacturer's requirements.
- Overhead seal oil tanks will be flushed by jumpering to compressor reference gas lines between compressor, overhead tanks and drainers are required to be flushed.

2.5 Stainless steel 100 mesh screens with back up 60 mesh screens with a number of spares must be fabricated with retaining gaskets and installed at selected lube oil piping flanges. This fabrication involves cutting and fitting the screen to the gaskets. Screens must be clearly and permanently tagged for ease of identification. Location for screens must be agreed with by vendor and owner. Basically, they should be positioned at all inlets to the machine. Locations immediately after risers must be avoided. Additionally, 100 mesh screens with 60 mesh back up screens, will be installed at the main oil return, degassing tank inlet and return and reservoir oil fill connection. These screens will be in place during all flushing operations. A drain should be fitted at lower point of return lines ahead of screen in order to deal with a blocked screen. Also, a pressure device (manometer-ie: simple length of plastic tubing) is to be installed at this point to monitor any pressure build-up due to blockage.

2.6 The reservoir shall be filled with the lube oil specified for permanent plant operation unless directed otherwise by the owner.

2.7 Lube oil flush should not occur until the compressor skid has been fully grouted.

2.8 The compressor rear bearing port cover and the load coupling guard must be installed in order to minimize any oil spill during flushing.

2.9 The vent piping must be checked out for mechanical completeness.

2.10 Check for correct operation of the auxiliary lube oil pump on/off/auto switch and the oil high temperature alarm before commencing flushing.

2.11 The instruments required for the flushing operation shall be identified on a P&ID mark-up for flushing and will be calibrated for normal operation prior to starting the oil system.

2.12 Add nitrogen bottles or instrument air connections at suitable tapping points downstream of lube and seal oil filters.

2.13 Water to oil coolers shall be provided.

2.14 An auxiliary boiler will be provided to heat the oil to approximately 180°F.

3. Flushing procedure

3.1 The range of temperatures for the hot oil circulation flush shall be 120°F to 180°F. Before initial circulation the oil should be heated to approximately 120°F.

3.2 The following parameters shall be documented in the log on an hourly basis:

- Pump discharge pressure

- Bearing header pressure
- Oil reservoir temperature
- Bearing header temperature
- Oil filter differential pressure
- Filter in use
- Sections of piping being flushed
- Start time

3.3 The drain oil sight flow gauges shall be continually monitored for flow at all places. The complete filling of the sight glass indicates a flow blockage. Immediate action shall be taken to stop circulation and clean filter screens. The debris obtained on the screen shall be collected into plastic bags, identified by screen location, machine number and time.

3.4 The following schedule shall be used for flushing:

- Add 100 mesh screen with 60 mesh backup screens at oil reservoir and degassing tank if applicable as stated in paragraph 2.5 and flush through total system at 15 minute intervals until screens are reasonably clean. Monitor closely the return lines on a continuous basis to ensure system is not backing up with oil.
- Flush through total system at intervals of one hour until screens are reasonably clean.
- Flush total system, alternating through systems section until screens are clean. Supply lines shall be alternated to ensure a greater than 150 percent oil flow is maintained at all times.
- Add 100 mesh screens with 60 mesh backup screens to lube and seal oil inlet lines to all equipment. Also add 100/60 mesh screens to overhead and seal oil tanks, reference lines and coupling guard feed lines (if furnished).
- Repeat 'flush total systems' above.
- Alternate flushing through each filter/cooler section, control valves and their bypasses, overhead tanks, seal oil traps and reference gas lines. *Record*

  Note: A differential pressure of 15 PSIG across either filter indicates the need for filter change.
- Bubble nitrogen through system at regular intervals. *Record*
- Flush through all instrument connections. *Record*
- Flush through all pressure control valve impulse lines. *Record*
- Thermoshock the system by use of the lube oil coolers at regular intervals (varying oil temperature between 120°F - 180°F). *Record*
- Rap exposed piping with a fibre hammer at one hour intervals. *Record*

3.5 When the 100 mesh screens meet criteria in paragraph 3.9/3.10 reinstate all instrumentation, orifices, pipe spools, etc. Arrange entire system in normal, complete configuration with all controls, alarms, etc., in operation.

3.6 Add 100 mesh white cloth (backed with 60 mesh screen) to all lube and seal oil supply points on equipment bearings, seals and control oil inlets. Flush until the criteria as stated in paragraph 3.9/3.10 is achieved, but for a minimum period of 24 hours.

Note: During final flush, alternate flushing through each filter/ cooler section, control valves and their bypasses.

3.7 Whenever practical, circulation shall be continued from the time of start-up until completion on a 24 hour per day basis. Any irregularities shall be immediately reported to the vendor representatives and the applicable owner representative as designated, which will be posted on the accessory skid control panel.

3.8 Owner's quality control representative will monitor all operations for compliance and verify all records, test parameters and acceptance criteria on a surveillance basis. Final screen particle count will be verified and recorded by quality control.

3.9 The oil system acceptance criteria which shall be the basis for witness approval parameters for contractor and/or owner shall be as follows:

Screen contamination shall be within the particle count limits according to the size of pipe it will be determined as follows:
20 Non-metallic particles on pipe 1" to 2"
50 Non-metallic particles on pipe 2" to 4".

3.10 Particles shall not be metallic and shall display random distribution on the screen.

3.11 The acceptance of the system shall be after installation/inspection 100 mesh cloth covered screens, then circulating the lube oil an additional four hours and re-inspecting the screens. The final four hours flush should be with valves open allowing full flow through the system at operating temperature. Final acceptance requires an oil analysis to determine metal content, viscosity and water content.

3.12 Following acceptance the system shall be restored by the owner for normal operation including the following actions:

- Remove all screens
- Visually inspect overhead seal oil tanks, degassing tanks, drainer modules, if required, wipe clean with lint-free cloth.
- Replace all components. Clean filter cartridges (less than 5PSI pressure drop) can remain subject to owner approval. If new cartridges are fitted, the cleanliness of the system must be re-checked.
- Restore reservoir oil to normal operating level.
- Obtain oil analysis.

System identification ____

Oil type ____________________ Req'd circulation temp ______________

Minimum duration req'd ____

Pre-flush checklist

		Owner	Vendor
1.	System configuration per reference drawings.	__________	__________
2.	Old oil has been removed and reservoir/filter housing wiped clean.	__________	__________
3.	Filters and screens installed for flushing.	__________	__________
4.	All required instrumentation calibrated.	__________	__________
5.	Lube/seal oil filled to capacity.	__________	__________
6.	Bearing housings ready to accept flow.	__________	__________
7.	Initial oil temperature reached.	__________	__________
8.	Approval to initiate oil flush.	__________	__________

Oil flush acceptance

The oil flush of the subject system has been conducted in accordance with site procedures and accepted.

Date of flush __________ Time started __________ Time completed __________

Inspected and accepted by __________ Date __________

Owner __________

Vendor representative __________

Oil Flush Log

Sheet __of __

System identification ____________ Inspector ____________

Date	Time	Circulating pump No.	Discharge pressure	Bearing/seal header		Reservoir temperature	Filter in circulation	Comments
				pressure	temperature			

Best practices for developing an effective site pre-commissioning program for rotating equipment

The following items are based on site pre-commissioning experience and essentially are 'best practices'.

1. Define participation by group – contractor pre-commissioning group, contractor engineering, user pre-commissioning group, user engineering, etc.
2. Define what equipment will be checked out (see preliminary list)
3. Define need for any training programs in basics of rotating equipment – such programs would be for machinists, millwrights etc., involved in venture pre-commissioning start up program
4. Determine *exactly* where contractor's responsibility ends and user's begins during pre-commissioning
5. Review list of ordered user test equipment and define personnel with experience to operate such equipment
6. Draft field start up test procedures for: pumps, general purpose drivers, compressors – centrifugal, compressors – reciprocating, special purpose drivers, lube oil systems and seal oil systems
7. Review contractor's pre-commissioning manuals to determine if the items noted below are included. If they are not, decide if a separate document is required to be issued by the appropriate group
    - Steam line blow down
        - Full flow
        - Direction of blow
        - Install blinds where required
        - Installation of targets – stainless steel, etc.
        - Limits of acceptance – marks on target
    - Check each unit for major piping connections (process) to assure no piping strains by:
        - Confirming all spring hangers, pipe supports and expansion joints are properly installed
        - Disconnect pipe from equipment – maximum offset allowable
        - Check equipment alignment by double dial indicator method before and after disconnecting major pipes 0 maximum change = 0.002" total
    - Thoroughly flush all lube and seal systems per job flushing procedure
    - Check all instrumentation and controls around rotating equipment to assure correct operation and calibration. As a minimum, the following should be accomplished
        - Correct calibration and operation of all transmitters and switches
        - Operational check of local panel
        - Calibration of all gauges

  - Correct action of all control valves
  - Calibration and operation of all transmitters
  - Correct calibration and operation of all steam turbine governors
  - Calibration of each vibration channel (probe, cable, monitor)
  - Calibration of each temperature monitor channel
  - Correct operation of steam turbine turning gears

- Final check of machinery alignment per job alignment procedure
- Confirmation of correct coupling shaft end separation
- Confirmation of correct pump minimum flow orifice size by checking actual installed orifice
- Pumps used during pre-commissioning for water flush must have mechanical seals removed, inspected and thoroughly cleaned prior to operation
- Large electric motors (1000 bhp and above) should be meggered
- Process lines should be blown down, at full flow if possible, be sure not to contaminate system. If blow down is not possible, following must be done:
  - Visually inspect lines (3 ft diameter and over) from knock out pot to machinery flange
  - Temporary strainers, per project specifications must be installed for lines 3 feet diameter and smaller and are preferred wherever possible
  - Internals of control valves in process lines (compressor recycle valves particularly) must be removed for preliminary runs of equipment
  - Prior to operation of any rotating equipment for the first time, scope and agenda for test run must be approved by user engineer.

Functional lube/seal system test procedure outline

Objective: To confirm proper functional operation of the entire system prior to equipment start-up

Procedure format: Detail each test requirement. Specifically note required functions/set points of each component. Record actual functions/set points and *all* modifications made.

Note: All testing to be performed *without* the unit in operation.

I Preparation
- A. Confirm proper oil type and reservoir level
- B. Confirm system flush is approved and *all* flushing screens are removed
- C. Confirm all system utilities are operational (air, water, steam, electrical)
- D. Any required temporary nitrogen supplies should be connected
- E. All instrumentation must be calibrated and control valves properly set
- F. Entire system must be properly vented

II Test procedure
- A. Oil Reservoir
  1. Confirm proper heater operation
  2. Check reservoir level switch and any other components (TI's, vent blowers, etc.)
- B. Main pump unit
  1. Acceptable pump and driver vibration
  2. Absence of cavitation
  3. Pump and driver acceptable bearing temperature
  4. Driver governor and safety checks (uncoupled) if driver is a steam turbine
- C. Auxiliary pump unit
  Same procedure as item B above.
- D. Relief valve set point and non-chatter check
- E. Operate main pump unit and confirm all pressures, differential pressures, temperatures and flows are as specified on the system schematic and/or Bill of material
- F. Confirm proper accumulator pre-charge (if applicable)
- G. Confirm proper set point annunciation and/or action of *all* pressure, differential pressure and temperature switches.
- H. Switch transfer valves from bank 'A' to bank 'B' and confirm pressure fluctuation does not actuate any switches.
- I. Trip main pump and confirm auxiliary pump starts without actuation of any trip valves or valve instability.
  Note: Pressure spike should be a minimum of 30% above any trip settings
- J. Repeat step I above but slowly reduce main pump speed (if steam turbine) and confirm proper operation
- K. Simulate maximum control oil transient flow requirement (if applicable) and confirm auxiliary pump does not start
- L. Start auxiliary pump, with main pump operating and confirm control valve

and/or relief valve stability

Note: Some systems are designed to *not* lift relief valves during two pump operation

III Corrective action

- A. Failure to meet any requirement in Section II requires corrective action and re-test
- B. Specifically note corrective action
- C. Sign-off procedure as acceptable to operate

Electro-hydraulic governor functional test procedure outline

Objective:	To confirm proper system functional operation prior to equipment start-up.
Procedure format:	Detail each test requirement. Specifically note required functions/set points. Record actual functions/set points and all modifications made. Note: All testing to be performed *without* the unit in operation

I Preparation

- Confirm all shut down contracts are in the normal condition
- Confirm all power supplies are on
- Secure necessary test equipment
  - Pressure sources (nitrogen bottles) for pressure simulation at transmitters
  - Frequency generators for simulating speed signals

II Test procedure

- Take required action to put system in 'run' mode
- Open trip and throttle valve *only after* insuring the main steam block valve is closed
- Simulate turbine start, slow roll and any start sequence 'hold' points up to minimum governor operating point
- Confirm proper operation of 'raise' and 'lower' speed buttons
- Connect external process signal inputs (one at a time) and confirm proper governor action to input signal variation
- Check overspeed override feature
- Confirm automatic transfer to and from backup governor 'position control' for each of the following cases:
  - Loss of main governor power supply
  - Zero external input signal
  - Failure of 'final driver' (internal governor component)
  - Zero speed inputs
- Confirm manual transfer to and from backup governor and 'emergency override'
- Check raise and lower speed controls while in backup governor mode
- Confirm governor shutdown (trip) operation under the following conditions:
  - Overspeed setting
  - Failure of both main and backup governor controls

III Corrective action

- Failure to meet any requirement in Section II requires corrective action and re-test
- Specifically note corrective action
- Sign off procedure as acceptable to operate

Steam turbine solo run functional test procedure outline

Objective:	To confirm acceptable mechanical operation of the steam turbine, governor system and safety (trip) system
Procedure format:	Detail each test requirement. Specifically note required test limits (Note: shop test data should be used to define acceptable limits). Record actual test values using appropriate instrumentation and note all modifications made.

I Preparation

- Confirm all auxiliary system tests are complete (governor, lube system etc.)
- Confirm all inlet steam lines have been cleaned and signed off
- Confirm all installed instrumentation is calibrated
- Secure *all* required instruments
  - Calibrated pressure and temperature gauges
  - Occiloscope(s)
  - Vector filters
  - Amplifier(s)
  - Spectrum analyzer
  - Tape recorder or information gathering module
- 'Walk' all steam inlet, extraction and exhaust lines. Confirm *all* spring hangers are released (unlocked) and safety valves are installed

II Test procedure

- Confirm *all* auxiliary systems are operational and at proper conditions (lined out)
  - Lube/control oil system
  - Governor/trip system
  - Turning gear (if applicable)
  - All warming lines drained and operational
  - Condensing system including condensate pumps (if applicable)
  - Extraction system (if applicable)
  - Steam seal system
  - Condition monitoring systems (vibration, temperature, etc.)
  - Steam conditions within allowable vendor limits
- Slow roll and start unit per vendors instructions. (Refer to cold start-up speed vs. time chart)
- Demonstrate manual trip (panic button) at low speed (500 rpm)
- Reset trip, accelerate back to desired speed – listen for rubs etc.
- Gradually increase speed to next speed step. Record the following data for each vendor required speed step up to minimum governor speed
  - Overall vibration at each vibration point (record frequency if specified limits are exceeded)
  - Bearing oil temperature rise at each bearing
  - Bearing pad temperature (axial and radial) at each point (if applicable)

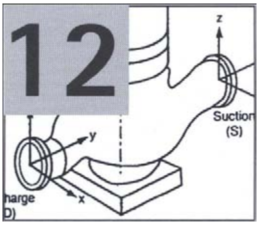
  - Turbine speed
  - Axial shaft displacement
  - Turbine exhaust temperature

  Note: Use shop test data for comparison

- After confirming stable operation at minimum governor speed, accelerate carefully to overspeed trip setting and trip the turbine three times. Each trip speed should fall within the vendors' trip speed set point allowable range.
- Return to minimum governor speed and confirm satisfactory manual and automatic speed control. Also confirm automatic transfer from main to backup governor.
- Connect vibration recording instruments, reduce turbine speed to 500 rpm and record shaft vibration (at each vibration monitoring point) and phase angle while gradually increasing speed to maximum continuous speed. Repeat step in reverse direction (maximum continuous speed to 500 rpm).
  Note: This data will be reduced to Bode , Nyquist and Cascade plots and should be compared to shop test data.
- Increase turbine speed to maximum continuous speed and run for four (4) hours or until bearing temperatures stabilize
- Finally trip the turbine using a system trip switch (simulate low oil pressure etc.)

III Corrective action

- Failure to meet any requirement in Section II requires corrective action and re-test
- Specifically note any corrective action
- Sign off equipment as acceptable to operate.



Pipe stress and soft foot effects on component failure

- Introduction
- How pipe stress and soft foot can cause component failure
- The root causes of excessive pipe stress and soft foot
- Condition monitoring indications of excessive pipe stress and soft foot
- Confirming excessive pipe stress and/or foundation forces (soft foot)
- Correcting excessive pipe stress and foundation forces on equipment
- Implementation of the action plan

Introduction

Have you ever been called into your supervisor's office and questioned on how to properly install the equipment or a component? Have you ever had the experience of installing a component (bearing) only to have it fail repeatedly over the following months and became a 'bad actor'? What is the problem? Your assembly procedures, the installation, the equipment or the process?

The subject of this chapter is equipment pipe stress and soft foot. Without a doubt, these factors are prime contributors to 'bad actors'. They both are relatively easy to prove; however, they can be very difficult to correct. The purpose of this chapter is to present the reasons why excessive pipe stress and soft foot cause bad actors, how to prove these problems exist and the most cost-effective method to correct them.

Before we can understand *how* pipe stress and soft foot can cause equipment component failures, we must know *what* pipe stress and soft foot are! Figure 12.1 presents these facts.

Pipe stress and soft foot exert failure producing forces on the equipment casing from:

- Top, side or bottom flanges (pipe loads)
- Support feet (soft foot)

Figure 12.1 Pipe stress and soft foot

Naturally, all equipment cases are designed to accommodate reasonable pipe loads and minimal load due to soft foot. However, Figure 12.2 shows what the equipment designer assumes in this regard.

External force design assumptions

- Under the limit on external pipe force (on assembly dwg)
- Under the limit on external pipe moments (on assembly dwg)
- All support feet are flat and in the same plane
- Foundation under all support feet has been leveled (shimmed if necessary with stainless steel shims)
- All external pipe(s) and support feet are properly connected

Figure 12.2 External force design assumptions

How pipe stress and soft foot can cause component failure

Figures 12.3 and 12.4 show a typical single stage overhung pump and a general purpose steam turbine respectively.

In both figures, the process pipes are not connected. If, in addition, both the pump and steam turbine were not coupled or bolted to their bases, what would cause the load (force) on the bearings?

Hopefully your answer was the rotor. Let's use the pump in the following discussion (Figure 12.3). However, everything discussed will equally apply to the steam turbine or any other type of equipment.

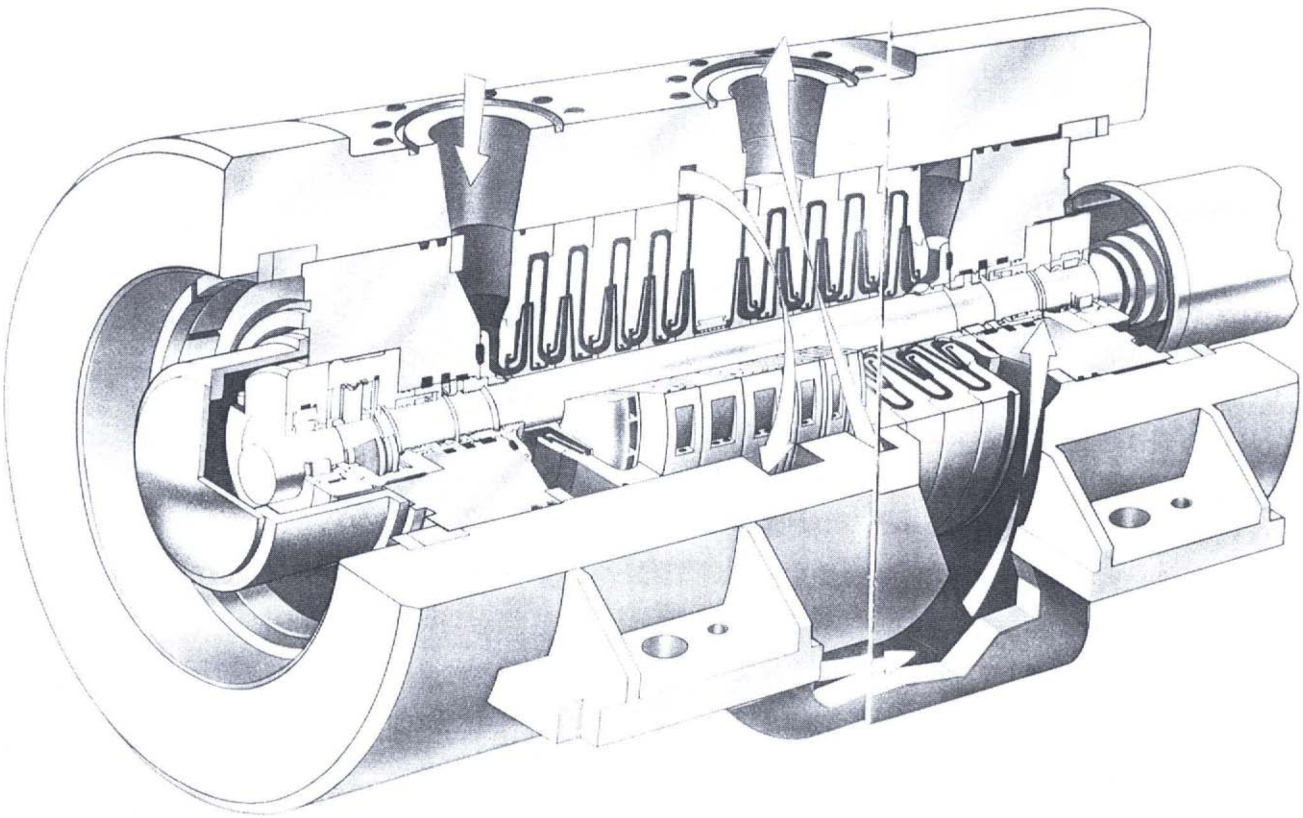


Figure 12.3 Single stage overhung pump (Courtesy of Union Pump Co)

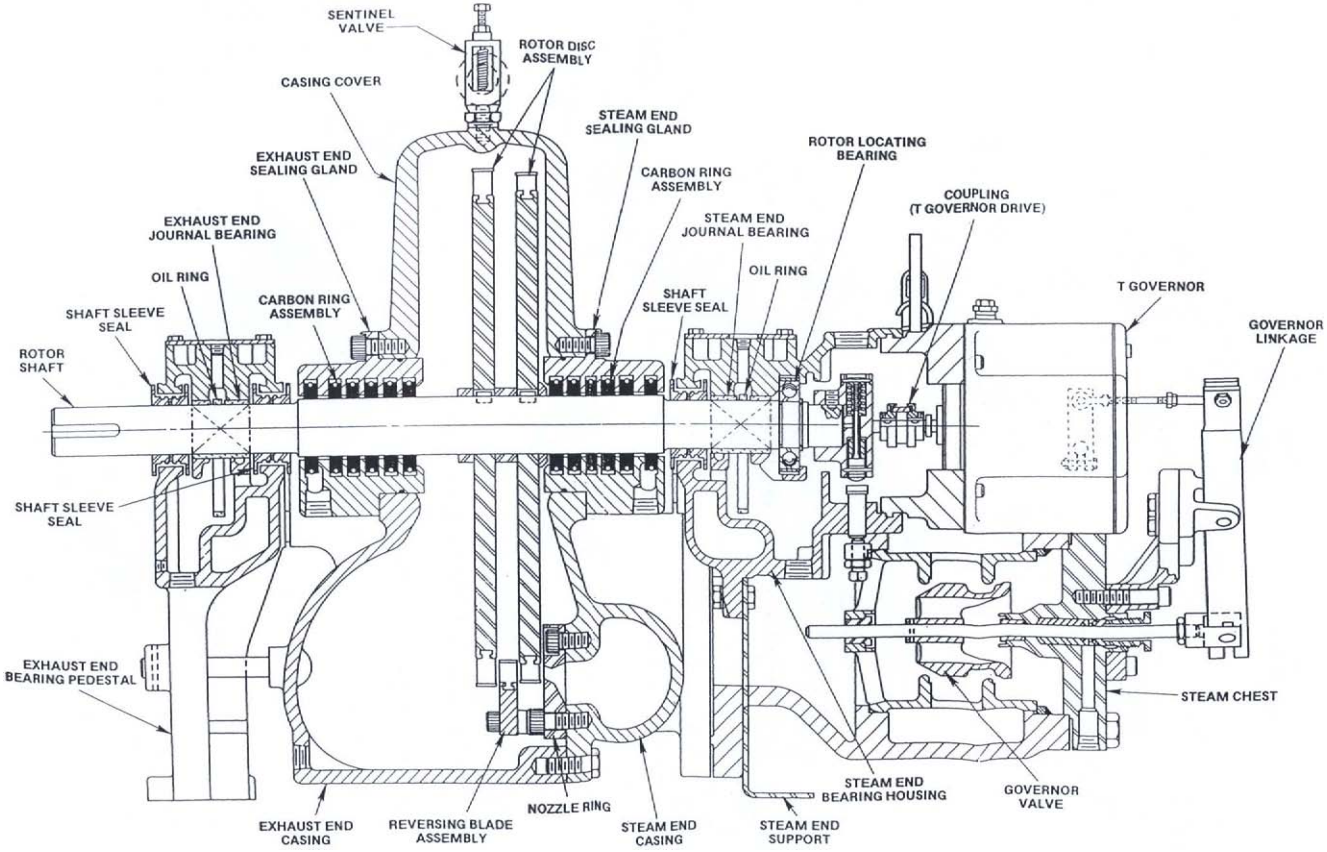

SENTINEL VALVE
ROTOR DISC ASSEMBLY
CASING COVER
STEAM END SEALING GLAND
ROTOR LOCATING BEARING
EXHAUST END SEALING GLAND
CARBON RING ASSEMBLY
COUPLING (T GOVERNOR DRIVE)
EXHAUST END JOURNAL BEARING
STEAM END JOURNAL BEARING
OIL RING
OIL RING
SHAFT SLEEVE SEAL
CARBON RING ASSEMBLY
SHAFT SLEEVE SEAL
T GOVERNOR
GOVERNOR LINKAGE
ROTOR SHAFT
SHAFT SLEEVE SEAL
EXHAUST END BEARING PEDESTAL
STEAM CHEST
STEAM END BEARING HOUSING
GOVERNOR VALVE
NOZZLE RING
EXHAUST END CASING
REVERSING BLADE ASSEMBLY
STEAM END CASING
STEAM END SUPPORT

Figure 12.4 General purpose steam turbine (Courtesy of Elliott Co)

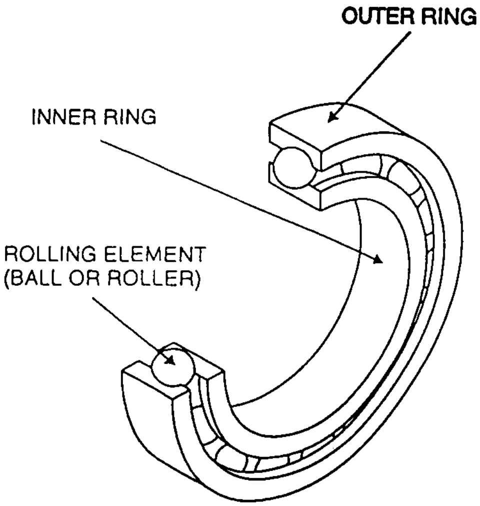


Figure 12.5 Radial bearing

Please refer to Figure 12.5, which shows a typical anti-friction bearing that would be used for the pump radial bearing.

Figure 12.6 presents the sources of the forces on any radial and/or thrust bearing regardless of the bearing type (anti-friction or sleeve).

Sources of forces

- Increased process pipe forces and moments
- Foundation forces ('soft' foot, differential settlement)
- Fouling or plugging of impeller
- Misalignment
- Unbalance
- Rubs
- Improper assembly clearances
- Thermal expansion of components (loss of cooling medium, excessive operating temperature)
- Radial forces (single volute – off design operation)
- Poor piping layouts (causing unequal flow distribution to the pump)

Figure 12.6 Sources of forces

For the pump in Figure 12.3, please describe the forces that the designer takes into account during the bearing *selection*. (Remember – anti-friction bearings are not custom designed). Circle the forces in Figure 12.6 that should be considered during the bearing selection.

Now please refer to Figure 12.7 which describes the relationship to determine the life of any anti-friction bearing.

'B' or 'L' – 10 life

'B' or 'L' - 10 life is defined as the life in hours that 9 out of 10 randomly selected bearings would exceed in a specific application.

$$\text{'B' or 'L' - 10 life} = \frac{16700}{N}\left[\frac{C}{F}\right]^3$$

Where: N = RPM

C = Load in lbs that will result in a bearing element life of 1,000,000 revolutions

F = Actual load in lbs

Figure 12.7 'B' or 'L' - 10 life

As an exercise, let's determine the bearing life for the following cases. Case 1 – no excessive external load. Case 2 – additional soft foot load. Case 3 – additional pipe stress load. We will note this information in Figure 12.8.

External loads on equipment example
(use the relationship in Figure 12.7)

Case	1	2	3
N (speed)	3600 rpm	3600 rpm	3600 rpm
C (bearing dynamic load factor – lbs)	3000	3000	3000
F (total actual bearing load – lbs)	170	500	1000
Condition	As designed	Additional soft foot forces	Additional pipe load forces
L-10 life years	25,495 hours	1002 hours	125 hours
Cause of early failure	No early failure specified minimum life was exceeded	Excessive soft foot forces	Excessive pipe load forces

Note: use the relationship in Figure 12.7 to determine the bearing L-10 life.

Figure 12.8 External loads on equipment example

In Figure 12.8, case 1 represents a bearing selected in accordance with industry standards that was installed correctly. That is, the predicted life of the bearing is in excess of 25,000 hours or 3 years continuous operation.

Cases 2 and 3 are a different story! Observe the dramatic effect of additional forces on the equipment casing from either soft foot or piping forces.

If your manager or an operator had to complete this exercise, he probably would have listed the 'machinist' as the cause of failure! Hopefully, this exercise has clearly demonstrated why components, especially bearings, can suddenly fail for no apparent reason. These facts are presented in Figure 12.9.

How excessive pipe stress and soft foot forces cause equipment component failure

- They exert forces in excess of design limits on the components
  - Casing
  - Bearings
  - Rotor
  - Seals
  - Wear rings
- The path of the forces is from the external force through the casing, bearing bracket to the components

Figure 12.9 How excessive pipe stress and soft foot forces cause equipment component failure

Figure 12.10 has been modified to show the force path from excessive discharge flange loadings to the bearing bracket.

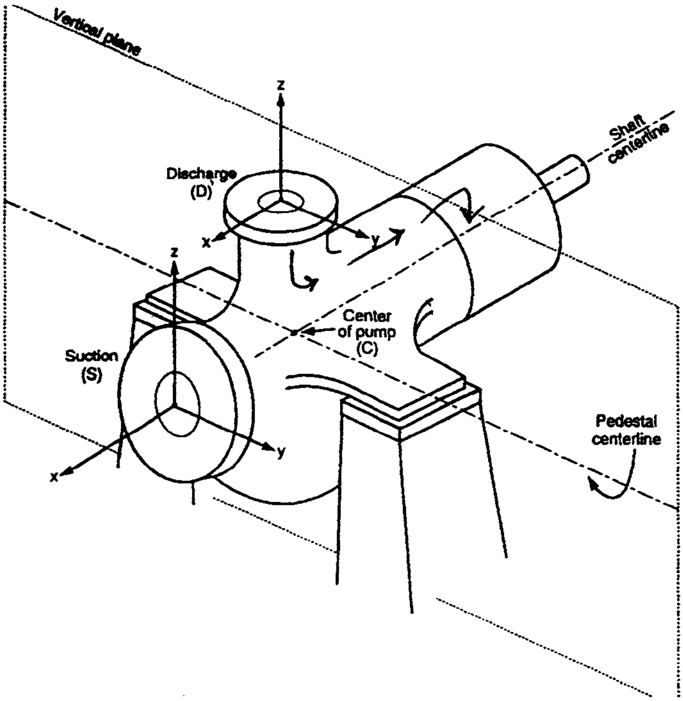

Vertical plane
z
Shaft centerline
Discharge (D)
x
y
z
Center of pump (C)
Suction (S)
y
x
Pedestal centerline

Figure 12.10 Force path from excessive discharge flange loadings to the bearing bracket (Courtesy of API)

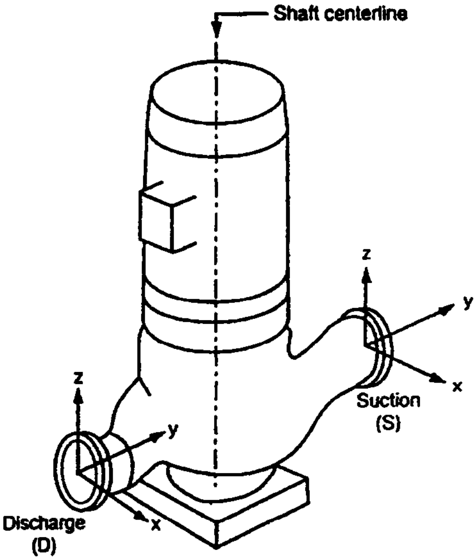

Shaft centerline
z
y
x
Suction
(S)
z
y
x
Discharge
(D)

Figure 12.11 Vertical in-line pump (Courtesy of API)

Figures 12.11 and 12.12 show the orientation of external flange forces and moments are referred to in the table shown in Figure 12.13.

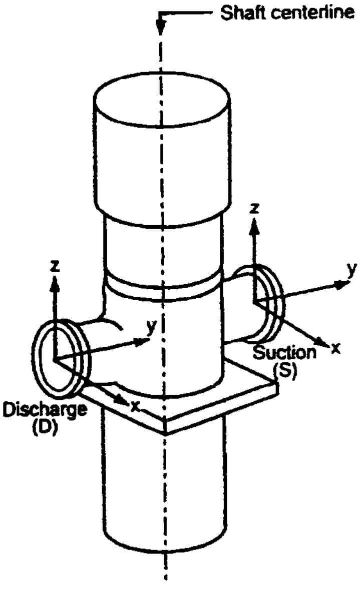

Shaft centerline
z
y
x
Suction
(S)
z
y
x
Discharge
(D)

Figure 12.12 Vertically suspended double-casing pump (Courtesy of API)

Table A – Nozzle loadings (SI units)

Note: each value shown below indicates a range from minus that value to plus that value; for example 710 indicates a range from −710 to +710

	Nominal size of flange (NPS)								
Force/moment	2	3	4	6	8	10	12	14	16
Each top nozzle									
FX	710	1070	1420	2490	3780	5340	6670	7120	8450
FY	580	890	1160	2050	3110	4450	5340	5780	6670
FZ	890	1330	1780	3110	4890	6670	8000	8900	10230
FR	1280	1930	2560	4480	6920	9630	11700	12780	14850
Each side nozzle									
FX	710	1070	1420	2490	3780	5340	6670	7120	8450
FY	890	1330	1780	3110	4890	6670	8000	8900	10230
FZ	580	890	1160	2050	3110	4450	5340	5780	6670
FR	1280	1930	2560	4480	6920	9630	11700	12780	14850
Each end nozzle									
FX	890	1330	1780	3110	4890	6670	8000	8900	10230
FY	710	1070	1420	2490	3780	5340	6670	7120	8450
FZ	580	890	1160	2050	3110	4450	5340	5780	6670
FR	1280	1930	2560	4480	6920	9630	11700	12780	14850
Each nozzle									
MX	460	950	1330	2300	3530	5020	6100	6370	7320
MY	230	470	680	1180	1760	2440	2980	3120	3660
MZ	350	720	1000	1760	2580	3800	4610	4750	5420
MR	620	1280	1800	3130	4710	6750	8210	8540	9820

Note 1: *F* = force in Newtons; *M* = moment in Newton meters; *R* = resultant. See Figures 12.11 and 12.12 for orientation of nozzle loads (X, Y and Z)
Note 2: Coordinate system has been changed from API Standard 610, 7th edition, convention to ISO 1503 convention.

Figure 12.13 Tables A and B – Nozzle loadings (Courtesy of API)

Table B – Nozzle loadings (US units)

Note: Each value shown below indicates a range from minus that value to plus that value; for example 160 indicates a range from −160 to +160.

	Nominal size of flange (NPS)								
Force/moment	2	3	4	6	8	10	12	14	16
Each top nozzle									
FX	160	240	320	560	850	1200	1500	1600	1900
FY	130	200	260	460	700	1000	1200	1300	1500
FZ	200	300	400	700	1100	1500	1800	2000	2300
FR	290	430	570	1010	1560	2200	2600	2900	3300
Each side nozzle									
FX	160	240	320	560	850	1200	1500	1600	1900
FY	200	300	400	700	1100	1500	1800	2000	2300
FZ	130	200	260	460	700	1000	1200	1300	1500
FR	290	430	570	1010	1560	2200	2600	2900	3300
Each end nozzle									
FX	200	300	400	700	1100	1500	1800	2000	2300
FY	160	240	320	560	850	1200	1500	1600	1900
FZ	130	200	260	460	700	1000	1200	1300	1500
FR	290	430	570	1010	1560	2200	2600	2900	3300
Each nozzle									
MX	340	700	980	1700	2600	3700	4500	4700	5400
MY	170	350	500	870	1300	1800	2200	2300	2700
MZ	260	530	740	1300	1900	2800	3400	3500	4000
MR	460	950	1330	2310	3500	5000	6100	6300	7200

Note 1: *F* = force in pounds; *M* = movement in foot pounds; *R* = resultant. See Figures 12.11 and 12.12 for orientation of nozzle loads (X, Y and Z)

Note 2: Coordinate system has been changed from API Standard 610, 7th edition, convention to ISO 1503 convention.

Figure 12.13 Continued – Tables A and B – Nozzle loadings (Courtesy of API)

Figure 12.13 shows that the allowable forces and moments for most pumps are very low!

The root causes of excessive pipe stress and soft foot

Refer to Figure 12.14, the machinery environment. An associate of mine has a favorite quote regarding the machinery environment.

The rotating equipment environment

- Process condition change
- Piping and foundation change
- 'Unit' (driven, driver, transmission, auxiliaries)
- Ambient conditions

Figure 12.14 The rotating equipment environment

'Stand at the equipment unit and rotate yourself 360°. Everything that you see can and will affect the reliability of this piece of machinery'.

As shown in the last section of this chapter, the cause of component failure is the excessive forces exerted on the equipment from:

- The piping
- The foundation (soft foot)

What then are the possible causes? There are many. We will divide the possible causes into the following categories:

- Design
- Construction
- Plant conditions

The possible root causes are presented in Figures 12.15, 12.16 and 12.17.

Possible causes for excessive pipe stress and/or soft foot (design)

- Pipe stress calculation error
- Improper spring hanger selection
- Improper soil analysis assumptions
- Improper foundation design

Figure 12.15 Possible causes for excessive pipe stress and/or soft foot (design)

Possible causes for excessive pipe stress and/or soft foot (construction)

- Using equipment as a 'pipe support'
- Improper installation of fixed spring supports
- Poor foundation and/or grout preparation
- Poor foundation and/or grout pour

Figure 12.16 Possible causes for excessive pipe stress and/or soft foot (construction)

Please refer to foundation and grout best practices presented in Chapter 11.

There have been numerous examples, especially in the Middle East, of poorly prepared foundations and grouting. Careful attention must be paid to the quality of the water used, the type of grout and the method of grouting followed.

It is strongly recommended that an epoxy grout be used for all rotating equipment and a grouting procedure, approved by a reputable epoxy grout manufacturer, be utilized.

Possible root causes for excessive pipe stress and/or soft foot (plant conditions)

- Settling pipe support foundations
- Cracked grout and/or foundation (concrete)
- Locked spring supports
- Improperly installed new spring supports
- Shim corrosion under pipe supports
- Shim corrosion between equipment feet and baseplate
- Shims vibrating loose under pipe supports

Figure 12.17 Possible root causes for excessive pipe stress and/or soft foot (plant conditions)

Figure 12.17 has an important message ... 'If you suspect excessive pipe stress and/or soft foot forces, get out and thoroughly walk around the affected machine'.

Condition monitoring indications of excessive pipe stress and soft foot

At this point, we have covered the function of the two most important components in rotating equipment:

- Bearings
- Seals

The components most commonly affected by excessive piping and/or foundation (soft foot) forces are the bearings and couplings, although seal reliability can be affected along with the bearings in extreme cases. Figures 12.18 and 12.19 present the parameters to monitor, as well as the limits for anti-friction and sleeve type radial bearings.

Condition monitoring parameters and their alarm limits

Bearing (anti-friction)

Parameter	**Limits**
1. Bearing housing vibration (peak)	.4 inch/sec (10 mm/sec)
2. Bearing housing temperature	185°F (85°C)
3. Lube oil viscosity	off spec 50%
4. Lube oil particle size	
■ non metallic	25 Microns
■ metallic	any magnetic particle in the sump
5. Lube oil water content	below 200 ppm

Figure 12.18 Condition monitoring parameters and their alarm limits

Condition monitoring parameters and their alarm limits

Bearing (hydrodynamic)

Parameter	Limits
1. Radial vibration (peak to peak)	2.5 mils (60 microns)
2. Bearing pad temperature	220°F (108°C)
3. Radial shaft position*	>30° change and/or 30% position change
4. Lube oil supply temperature	140°F (60°C)
5. Lube oil drain temperature	190°F (90°C)
6. Lube oil viscosity	off spec 50%
7. Lube oil particle size	>25 microns
8. Lube oil water content	below 200 ppm

* except for gearboxes where greater values are normal from unloaded to loaded

Figure 12.19 Condition monitoring parameters and their alarm limits

As previously stated, excessive pipe strain and soft foot exert forces beyond the design limits on equipment components. In the case of bearings, the forces will be transmitted from the source (pipe and/or foundation) through the casing to the bearing housing, to the bearing. Figure 12.20 presents the facts concerning how to determine by condition monitoring if we have a pipe stress and/or soft foot problem.

Condition monitoring indications of excessive pipe forces and/or soft foot

- More than one (1) bearing failure, rotor breakage, or coupling failure per year
- Unexplained high vibration (usually indicating misalignment)
- Unexplained high bearing housing temperature
- Pipe supports close to equipment *not* vibrating, equipment is vibrating

Figure 12.20 Condition monitoring indications of excessive pipe forces and/or soft foot

Confirming excessive pipe stress and/or foundation forces (soft foot)

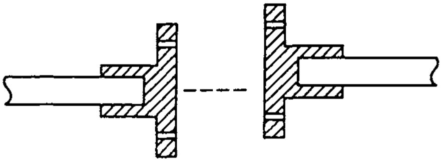
Once the root causes of machinery component failure are suspected, they must be confirmed. Once confirmed, a cost-effective action plan must be developed that will assure implementation at the earliest opportunity. This section presents this important information.

Confirming excessive pipe stress and/or foundation forces

In order to implement any action, we had better be sure our suspected root causes are correct. If they are not, we will always have a difficult time obtaining approval for any future recommendation.

Figures 12.21 and 12.22 present the guidelines for confirmation of excessive piping stress and/or soft foot.

Shaft alignment

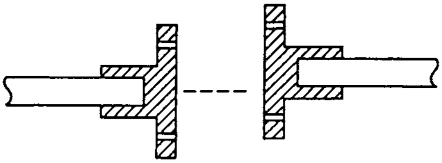


Preliminary considerations

1. Obtain thermal shafts and machine growth calc's establish 'cold offsets'
2. Coupling hubs installed in accordance with OEM's procedures
3. Set proper B.S.E. dimension
4. Test for 'soft foot' 0.002 maximum allowable differential rise
5. Use only stainless shims – minimize number of shims
6. Shims must straddle hold down bolts

Figure 12.21 Shaft alignment – Preliminary considerations

Shaft alignment



Alignment change limits when connecting piping

1. Mount dial indicators independent of machinery horizontally and vertically reading on coupling flanges
2. Zero out indicators
3. Tighten using specified sequence and torque values
4. Maximum shaft movement = 0.002
5. Dowell if required by OEM

Figure 12.22 Shaft alignment - Alignment change limits when connecting piping

Correcting excessive pipe stress and foundation forces on equipment

Of all the different problems with rotating equipment, the resolution of excessive pipe stress is the most difficult.

Why? Correction can involve extensive work that will require a significant amount of safety permits and may even require process unit shutdown.

Figure 12.23 shows the suggested excessive pipe stress solution procedure.

It is naturally arranged in a cost-effective order (simplest, least costly action first).

Suggested excess pipe stress solution procedure

I Confirm excessive pipe stress[1] (refer to Figure 12.24). Also confirm pipe bolting can be removed without a 'come along'

II Walk piping system and confirm proper installation per piping isometrics

- Proper pipe support shims
- Spring supports free to move
- No obvious pipe misalignment

III Correct excessive pipe stress by:[2]

- Attempting rebolting at the next flange
- Using 'Dutchman' with flexitallic gaskets (each side)
- Heating of pipe for alignment
- Pipe modification at next T&I

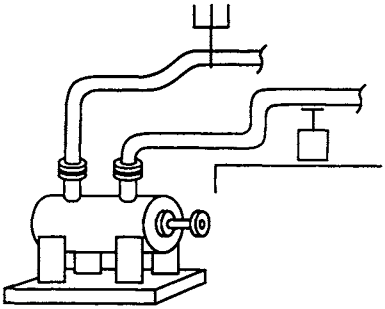
Notes: 1 Work permits required

2 All items in III must be confirmed correct per Figure 12.24.

Figure 12.23 Suggested excess pipe stress solution procedure

Pipe considerations

1. *Float pipe* to machine. Machine *is not* a pipe support
2. Lock pins on spring or pipe supports adjacent to machinery kept in place until system filled
3. Flanges parallel on machined surfaces and no come alongs! (Bolts can be removed without excessive force)



Flange faces within 0.0010"

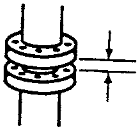


Figure 12.24 Piping considerations

Correcting soft foot problems can be extremely simple if equipment support feet are not level to the foundation. In this case, stainless steel shims can be added.

However, in some cases, baseplates can become distorted and/or the foundation can experience differential settlement over a period of time. This case usually requires that a complete new foundation be designed and installed at the next T&I. A short-term fix can be to install stainless steel shims to temporarily correct the problem.

Implementation of the action plan

Correction of pipe stress problems is usually the most difficult problem to obtain action plan implementation for.

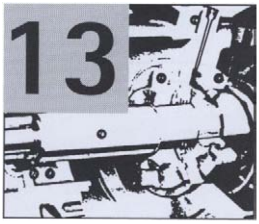
Why? It is costly, exposes the plant to possible safety problems and can result in a process unit shutdown. Usually, it should be planned for a T&I.

I have found that the guidelines in Figure 12.25 provide the best probability of implementation.

Obtain and maintain management support by ...

1. Clearly stating impact of problem on plant profit
2. Prepare a brief statement of:
   - The problem
   - Action plan and confirmation of success (past experience)
   - Cost of failure to date
   - Cost of solution
   - The impact on plant profit (loss)
3. Be confident!
4. Be professional!
5. Provide timely updates and final report on completion

Figure 12.25 Obtain and maintain management support by ...



The effects of misalignment on reliability

- Introduction
- Why misalignment reduces rotating equipment reliability
- How misalignment effects can be detected
- Alignment methods and guidelines

Introduction

Misalignment, as a cause of excessive machinery vibration, accounts for approximately 7 out of 10 cases! Without a doubt, it is the most common cause of a change in vibration. As we already know, increased vibration means increased forces on machinery components and therefore reduced machinery reliability. The subject of this chapter is therefore the effects of misalignment on equipment and component reliability.

Figure 13.1 presents sketches representing perfect alignment and the different types of misalignment.

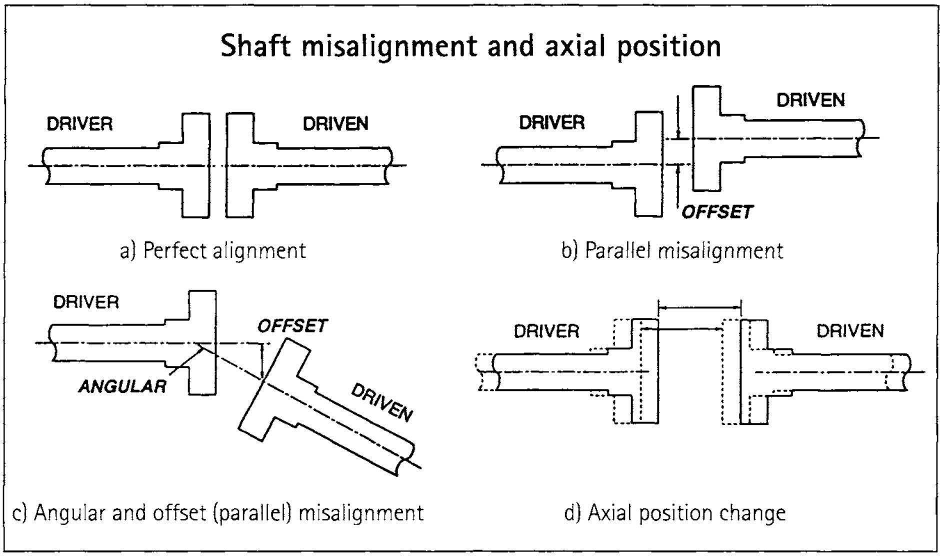

Shaft misalignment and axial position
DRIVER
DRIVEN
a) Perfect alignment
DRIVER
DRIVEN
OFFSET
b) Parallel misalignment
DRIVER
OFFSET
ANGULAR
DRIVEN
c) Angular and offset (parallel) misalignment
DRIVER
DRIVEN
d) Axial position change

Figure 13.1 Shaft misalignment and axial position

Perfect alignment occurs when centerlines of each mating shaft are the same in each plane X, Y and Z. This is shown in item a of Figure 13.1. Note: Figure 13.1 only shows X and Y planes.

Figures 13.1b, 13.1c and 13.1d show other types of misalignment, including axial misalignment in Figure 13.1d. All types of misalignment can occur simultaneously and usually do! Having presented the types of misalignment, let's now examine how misalignment can occur. For this discussion, we will use a common example of a motor driven centrifugal pump in Figure 13.2.

The categories of misalignment causes are shown in Figure 13.3.

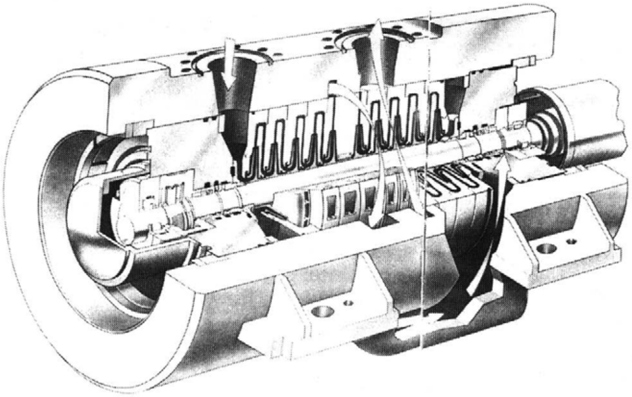


Figure 13.2 Motor driven overhung pump

Misalignment cause categories

- Cold 'offset' (vendor requirement)
- Environment induced
- Installation error induced

Figure 13.3 Misalignment cause categories

We will discuss each category in Figure 13.3 in detail.

Cold offset

Cold offset misalignment is the result of the alignment procedure. It is a misalignment set on purpose and dictated by the equipment vendor to account for the different values of equipment support growth from the 'cold' to hot running position. The values for the 'cold offset' are calculated by the equipment vendor using the standard equation for differential thermal expansion and assuming an operating temperature of the equipment support. Definition of terms and this relationship are presented in Figure 13.4. Please also refer to Figure 13.2 at this point.

Cold offset misalignment – definition of terms

- 'Cold offset misalignment' – stated by vendor in instruction book to account for shaft position change from 'cold' to 'hot' position.
- 'Cold position'* – position of shafts at rated ambient temperature
- 'Hot position'* – position of shafts at rated operating temperature of each equipment piece
- Thermal expansion of equipment support legs determined by:

  $\Delta L = (0.0000065)(L)\ (\Delta T)$ where:

  ΔL = thermal growth in inches

  0.0000065 = thermal constant for expansion of steel in inches per inch per °F

  L = original length of equipment support at ambient temperature – inches

ΔT = change of support temperature from 'cold' to 'hot' condition in °F

* Assumes acceptable pipe and soft foot forces as defined in previous chapter ('pipe stress and soft foot')

Figure 13.4 Cold offset misalignment – definition of terms

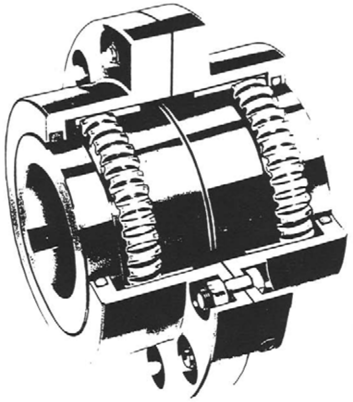


Figure 13.5 Gear tooth coupling (grease packed) (Courtesy of Zurn Industries)

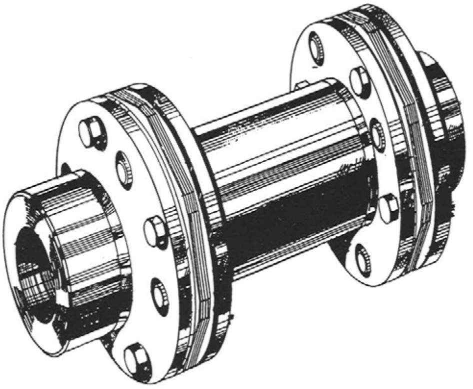


Figure 13.6 Flexible disc spacer coupling (Courtesy of Rexnord)

We can safely state at this point that 9 out of 10 'cold offset' values dictated by the vendor do not result in 'perfect shaft alignment'.

Please refer again to Figure 13.1 for the definition of perfect shaft alignment. It can be seen that perfect shaft alignment would minimize forces transmitted to the equipment bearings when equipment is connected (coupled). Naturally, a coupling must be utilized to transmit the power (torque) from the driver (motor, steam turbine, diesel engine, etc.) to the driven equipment (pump, gearbox, fan, etc.). All flexible couplings are designed to transmit torque while allowing for a certain amount of shaft misalignment (refer to Figures 13.1b, 13.1c and 13.1d). The two major classifications of flexible couplings are presented in Figures 13.5 and 13.6.

Present company specifications require the use of flexible disc couplings because they do not require lubrication as all gear couplings do. Depending on the operating speed, gear couplings must either be greased (operating speed below 4000 rpm) or continuously lubricated (operating speed above 4000 rpm). Failure to grease couplings or the accumulation of 'oil sludge' in continuously lubed couplings can cause the couplings to become 'non-flexible' and 'lock up'. This action can result in high vibration, radial bearing, thrust bearing, coupling and shaft failure. These facts are presented in Figure 13.7.

Choice of coupling classification

- Dry, non-lubricated couplings are required for all new applications (since approximately 1980) because:
  - They do not require lubrication
  - They do not 'lock up'
  - They do not transmit excessive forces to the equipment and components if:
    - The allowable maximum misalignment is not exceeded (approximately $^{1}/_{4}^{\circ}$)*
    - The axial expansion limit is not exceeded*

* Refer to coupling drawing for exact limits

Figure 13.7 Choice of coupling classification

As shown in Figure 13.7, a flexible coupling will allow the equipment to be installed and operated with some residual misalignment with 'cold offset' as long as the allowable limits are not exceeded.

An example shown in Figure 13.8 will demonstrate these facts. Please refer to Figure 13.2 again for this example.

Cold offset example

1. Task: calculate 'assumed' (vendor value) and actual shaft position for motor and pump in Figure 13.2 and data given below (use relationship in Figure 13.4).
2. Given data
   - Pump support length (centerline) to foot = 16"
   - Pump assumed average support temperature (rated condition) = 100°F
   - Pump actual average support temperature = 110°F
   - Motor support length (centerline) to foot = 12"
   - Motor assumed average support temperature = 125°F
   - Motor actual average support temperature = 115°F
   - Ambient temperature = 90°F
3. Calculation table (use Figure 13.4)

Parameter	L	ΔT Avg. °F	ΔL Vertical
Pump (assumed ΔT)	16	10°F	0.001
Pump (actual ΔT)	16	20°F	0.002
Motor (assumed ΔT)	12	35°F	0.003
Motor (actual ΔT)	12	25°F	0.002

4. What would be the vendor specified vertical offset? – Motor set 0.002" low
5. Based on the actual average support temperatures, what is the vertical misalignment? – 0.002"
6. Will this misalignment increase equipment bearing bracket vibration? – No
7. Why? – Because the misalignment is significantly less than the coupling misalignment limit

Figure 13.8 Cold offset example

Figure 13.8, when completed, will demonstrate a typical fact concerning vendor specified 'cold offset' – it does not usually result in perfect hot alignment!

Figure 13.9 presents guidelines for correcting misalignment resulting from improper 'cold offset' values.

Improper 'cold offset' values – corrective action

- If measured vibration at 'hot' operating conditions (after approximately 4 hours) exceeds limit (0.18 inch per second):
  - Measure temperatures of supports using accurate temperature instrument
  - Use average actual temperatures to calculate actual growth (refer to Figure 13.8)
  - Revise 'cold offset' values based on actual measurements

Figure 13.9 Improper 'cold offset' values – corrective action

Environment induced misalignment

Using the information gained in this book concerning process condition changes, bearings, lubrication and external piping and foundation forces, it becomes easy to understand that perfect alignment is not always possible. Figure 13.10 contains some of the machinery environment factors that can and will cause misalignment. Please refer to the previously presented figure on the rotating equipment environment and note all of the possibilities. We have included this information for your convenience in Figure 13.11.

Environmental factors affecting machinery alignment

- Ambient temperature change (influenced by: wind and weather conditions)
- Foundation stability (cracking, tidal changes, differential settlement, loose bolts)
- Induced piping forces (process condition changes, spring supports loose shims)
- Load induced equipment operating temperature change
- Lubrication system condition changes

Figure 13.10 Environmental factors affecting machinery alignment

The rotating equipment environment

- Process condition change
- Piping and foundation change
- 'Unit' (driven, driver, transmission, auxiliaries)
- Ambient conditions

Figure 13.11 The rotating equipment environment

A recommended corrective action plan, in the event of suspected environment caused misalignment, is presented in Figure 13.12.

Suspected environment caused misalignment – corrective action

- If vibration values and signature change indicate misalignment, check:
  1. Walk process piping - check for problems
  2. Check for foundation cracks and soft foot
  3. Trend vibration against the following:
     - ▲ Temperature of support feet
     - ▲ Process condition changes (flow, motor amps, steam conditions)
     - ▲ Bearing bracket temperature
     - ▲ Lube oil condition (oil level, water content, viscosity)

Note: Check values in item 3 against baseline (start up) values if possible.

Figure 13.12 Suspected environment caused misalignment – corrective action

Installation error induced misalignment

Whether we want to admit it or not, we all make errors at one time or another. And, it is usually when we are trying to complete a task against a deadline. Installation induced errors can range from incorrect measurements to simply forgetting to install the specified amount and size of shims. Please refer to Figure 13.13.

Alignment checklist

I Rough alignment
- Foundation free of voids (test by tapping for solid sound)
- Equipment properly leveled
- Stainless steel shims used
- All hold down bolts properly torqued
- Soft foot check acceptable
- Equipment aligned per site procedure using vendor 'cold offset' values

II Final alignment
- Re-check rough alignment values
- Re-check soft foot
- Install piping using plant procedures, vendor's instructions and guidelines presented in this workshop
- Install coupling in accordance with vendor's coupling drawing
  Notes: 1. Do not exceed recommended BSE (between shaft ends) dimension
  2. Do not use more shims than recommended by coupling vendor
- Confirm all piping to machinery bolting is properly torqued

Figure 13.13 Alignment checklist

Why misalignment reduces rotating equipment reliability

Simply stated, misalignment reduces rotating equipment reliability by exerting excessive external forces on all major machinery components. These forces can be excessive static loads and/or dynamic loads (causing changes in vibration). Figure 13.14 shows the machinery components that are affected by misalignment.

Machine components affected by misalignment

- Coupling
- Shaft
- Bearings (radial and thrust)
- Shaft end seals
- Internal seals (wear rings, etc.)
- Motor rotor and stator

Figure 13.14 Machine components affected by misalignment

In order to show how the reliability of the components in Figure 13.14 can be affected by misalignment, it is necessary to understand the mechanism of shaft misalignment.

Simple description of misalignment mechanism

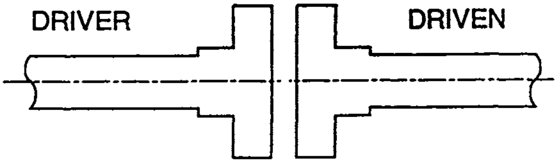
The forces generated by misalignment of mating shafts are the result of the centerline of shafts not being perfectly aligned. If the shafts were perfectly aligned, the trace of the shaft centerlines of the mating shafts for one revolution would be a point (see Figure 13.15a) (This assumes perfect balance and no other forces acting on either shaft).

If parallel misalignment existed (see Figure 13.15b), the trace of the shaft centerlines for one rotation would be two points equally spaced around the shaft centerlines.

Therefore, for each shaft resolution there would be both a + and – value of the offset. Note: For these discussions, we are assuming that the offset is in the vertical plane only. However, it can be in either of three planes (X, Y and Z). For the case of pure parallel misalignment, the forces are only in the radial direction and occur two times per revolution. Therefore, vibration will result at a frequency of two times per revolution. For pure parallel misalignment, axial forces are not present and axial vibration will remain unchanged if misalignment occurs.

The most common form of misalignment, angular and parallel misalignment, is shown in Figure 13.15c. In this case, the trace of the centerlines would again be two points equally spaced around the shaft centerlines *but* in a plane of rotation at some angle off the vertical plane. This would also result in additional axial vibration at a frequency equal to the rotational frequency. These facts are summarized in Figure 13.15.

The misalignment mechanism



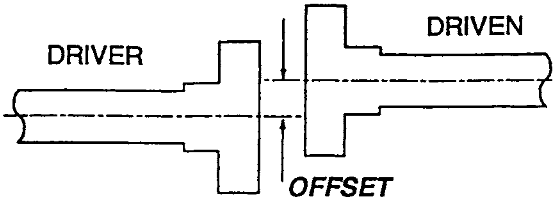
Perfect alignment

Facts:

- Trace of shaft centerline is a point at the shaft centerlines
- No additional radial or axial forces transmitted to components

Figure 13.15a The misalignment mechanism – Perfect alignment

The misalignment mechanism



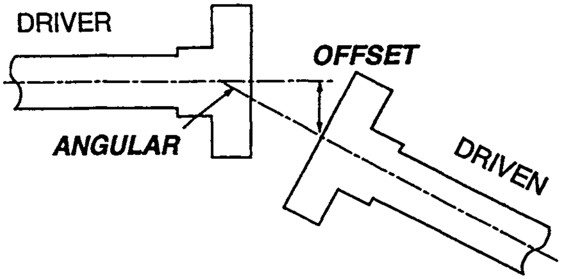
Parallel misalignment

Facts:

- Trace of shaft centerlines is two points equally spaced around the shaft centerlines
- The displacement of the offset shaft would be +/– the offset value around the shaft centerline
- The frequency of the vibration would be 2 x rotational speed
- Only radial vibration would exist; there would not be axial vibration

Figure 13.15b The misalignment mechanism – Parallel misalignment

The misalignment mechanism



Angular and offset (parallel) misalignment

Facts:

- Most common type of misalignment
- Trace of shaft centerlines is again two points equally spaced around the shaft centerlines
- Frequency of vibration again is 2 x rotational speed
- Since plane of affected shaft is not vertical, axial vibration at 1 x shaft rotation also exists

Notes: For Figures 13.15a, 13.15b and 13.15c

1. Only vertical offset is assumed
2. Only force present is misalignment force

Figure 13.15c The misalignment mechanism - Angular and offset (parallel) misalignment

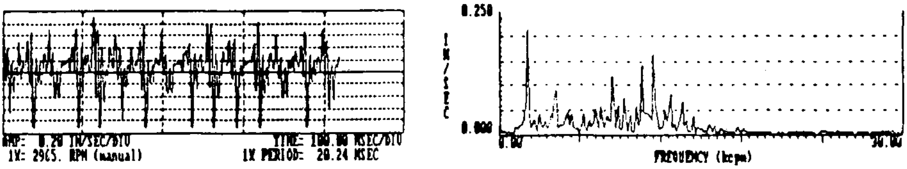
Induced forces and deflections

As demonstrated in Figure 13.15, all types of misalignment produce vibration and deflections because the forces acting on all components are dynamic (their load point is changing with shaft rotation). The frequency of vibration depends on the type of misalignment.

At this point, a brief explanation of 'overall vibration' and 'frequency' of vibration is necessary. Figure 13.16 is an actual overall vibration signal (unfiltered) taken from a pump in cavitation. Since the overall vibration value was above the normal limit, a filtered plot showing the various vibration frequencies contained in the overall (unfiltered) vibration signal was obtained.

Vibration frequency

Any vibration signal is made up of one or more frequencies. A typical unfiltered (overall) and filtered signal are shown below:



Unfiltered

Filtered

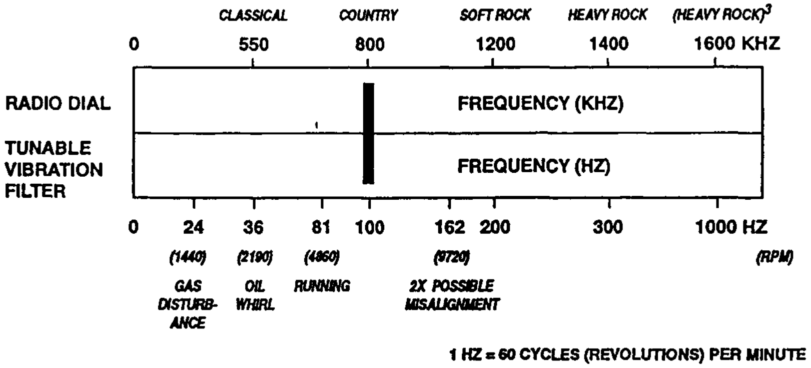
An analogy to a filtered signal is a radio. In a given locality, many stations are transmitting simultaneously. Any given station is obtained by adjusting the tuner to the correct transmission frequency.

Figure 13.16 Vibration frequency

The easiest way to understand vibration frequency is to use an analogy.

Figure 13.17 presents an example using a radio analogy courtesy of Charles Jackson, vibration consultant.

Radio tuner/vibration filter analogy



CLASSICAL
COUNTRY
SOFT ROCK
HEAVY ROCK
(HEAVY ROCK)^3
0
550
800
1200
1400
1600 KHZ
RADIO DIAL
FREQUENCY (KHZ)
TUNABLE VIBRATION FILTER
FREQUENCY (HZ)
0
24
36
81
100
162
200
300
1000 HZ
(1440)
(2190)
(4860)
(9720)
(RPM)
GAS DISTURB-ANCE
OIL WHIRL
RUNNING
2X POSSIBLE MISALIGNMENT
1 HZ = 60 CYCLES (REVOLUTIONS) PER MINUTE

By adjusting the tuner (filter) to a selected station (frequency) the desired program (vibration frequency signal) can be obtained if it is 'on the air' (present)

Figure 13.17 Radio tuner/vibration filter analogy (Courtesy of Charles Jackson)

Vibration frequency analysis is used to determine the possible cause(s) of any vibration change. Note that we did not say 'vibration increase' but 'vibration change'. Reduced vibration signifies reduced dynamic forces on the components but not necessarily total forces. In order to understand how excessive misalignment transmits forces to machinery components and reduces component life and reliability, please refer to Figure 13.18.

Misalignment can reduce machinery component life by:

Exerting excessive static (steady) and dynamic (vibration) forces on any or all of the following components

- Radial bearing
- Coupling
- Thrust bearing
- Shaft
- Seals

Figure 13.18 Misalignment can reduce machinery component life

To demonstrate this fact, we have included an example of an anti-friction bearing failure (the most common misalignment caused component failure) in Figure 13.19.

Misalignment caused radial anti-friction bearing failure

I Given: anti-friction bearing data

- Equipment type - centrifugal pump (Figure 13.20)
- L-10 Life = 25,000 hours (3 years)
- Shaft speed = 3600 rpm
- Dynamic load factor 'C' = 3000 lbs
- Total bearing force 'F' before misalignment = 100 lbs
- Total bearing force 'F' with excessive misalignment = 200 lbs (due to increased 'dynamic vibration')

II Find: radial bearing life

- without misalignment
- with excessive misalignment

Note: excessive misalignment is defined as that value to increase vibration beyond acceptable field limits

III Anti-friction bearing life relationship:

$$\text{L-10 (hours)} = \frac{16700}{\text{rpm}} \left[\frac{C}{F}\right]^3$$

A. 'No misalignment'

$$125{,}250 \text{ hours} = \frac{16700}{3600} \left[\frac{3000}{100}\right]^3$$

B. 'Excessive misalignment'

$$15{,}656 = \frac{16700}{3600} \left[\frac{3000}{200}\right]^3$$

IV Conclusions:

A. Do the conditions in case B cause this pump to be a 'bad actor'? - No

B. Why? - Because the additional 100 pounds force result in a bearing life that will cause failure approximately once every 2 years and a bad actor is defined as a component that requires replacement more than once a year.

Figure 13.19 Misalignment caused radial anti-friction bearing failure

As previously stated in this section, excessive misalignment can cause reduced life and failure of any component. Refer back to Figure 13.1d of this chapter and note that improper axial alignment can also be the root cause of coupling failure.

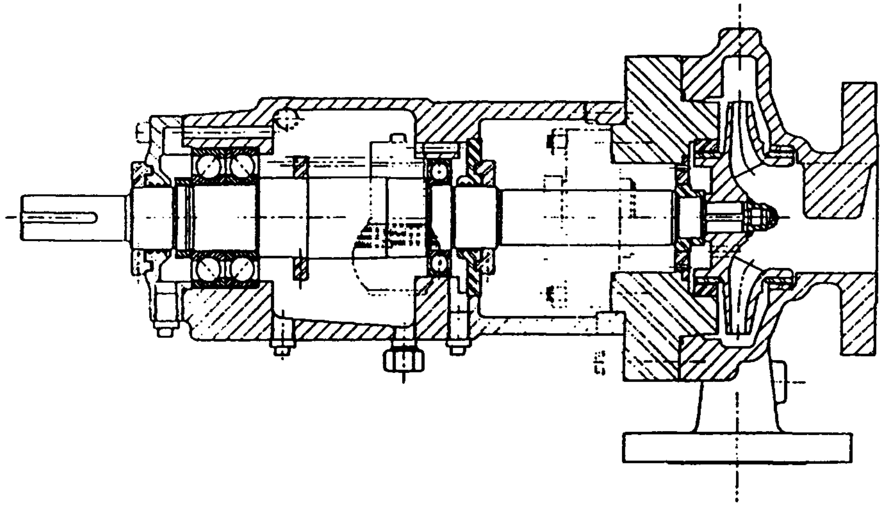


Figure 13.20

Dry, disc type couplings have very limited axial movement capability compared to gear type couplings (approximately 0.062" vs. 0.250"). Therefore, always carefully install dry, disc type couplings in complete accordance with vendor's instructions.

How misalignment effects can be detected

As we have learned throughout this book, monitoring the condition of all major components is the key to success. As we have also learned, the components most affected by excessive misalignment are bearings and couplings.

Bearing condition parameters

Unfortunately, it is not possible to monitor coupling condition directly. However, since the coupling is mounted close to the bearings and its balanced condition contributes to the bearing total force, bearing component condition monitoring is the single most important method of detecting misalignment changes. We have included in Figures 13.21 and 13.22 again the condition monitoring parameters and their limits for radial anti-friction and sleeve bearings.

Condition monitoring parameters and their alarm limits

Bearing (anti-friction)

Parameter	Limits
1. Bearing housing vibration (peak)	.4 inch/sec (10 mm/sec)
2. Bearing housing temperature	185°F (85°C)
3. Lube oil viscosity	off spec 50%
4. Lube oil particle size	
■ non metallic	25 microns
■ metallic	any magnetic particle in the sump
5. Lube oil water content	below 200 ppm

Figure 13.21 Condition monitoring parameters and their alarm limits

Condition monitoring parameters and their alarm limits

Bearing (hydrodynamic)

Parameter	Limits
1. Radial vibration (peak to peak)	2.5 mils (60 microns)
2. Bearing pad temperature	220°F (108°C)
3. Radial shaft position*	>30° change and/or 30% position change
4. Lube oil supply temperature	140°F (60°C)
5. Lube oil drain temperature	190°F (90°C)
6. Lube oil viscosity	off spec 50%
7. Lube oil particle size	>25 microns
8. Lube oil water content	below 200 ppm

* except for gearboxes where greater values are normal from unloaded to loaded

Figure 13.22 Condition monitoring parameters and their alarm limits

The two major items to observe for determination of excessive misalignment are:

- Vibration characteristics
- Bearing bracket or bearing pad (where applicable) temperature

It is important to note that measurement of vibration characteristics do not conclusively prove bearing overload. As noted in the example problem in the previous section, vibration is the result of dynamic load. I have seen cases of high bearing loads caused by excessive misalignment where vibration is actually decreased, but bearing temperature is increased. Remember, vibration measurements do not indicate the total bearing load! Bearing bracket and bearing pad temperature increase are the best indication of increased total bearing load. Note: If hydrodynamic bearings are used radial shaft centerline position change can also be used to confirm bearing load change.

Vibration measurement and analysis

This information was presented in the previous section since it was necessary for the understanding of how misalignment can reduce rotating equipment reliability.

Therefore, please refer back to Figures 13.16 and 13.17 for this discussion.

To determine if misalignment is a contributing cause for the failure of machinery components, refer to the guidelines presented in Figure 13.23 and refer to the contents of Figures 13.16 and 13.17.

Using vibration data to confirm misalignment (guidelines)

1. Compare baseline (start up) frequency scan with present scan for radial and axial readings
2. Observe changes in the following frequencies:
   - Radial 2X
   - Axial 1X
3. If overall (total) vibration values (radial and/or axial) change and either of the frequencies noted in item 2 increase, shut equipment down at first opportunity and check alignment.

Figure 13.23 Using vibration data to confirm misalignment (guidelines)

Alignment methods and guidelines

The intent of this section is not to conduct a 'clinic' on alignment. The intent is to provide 'best practice' information, based on experience concerning alignment methods and guidelines that will result in final shaft alignment values that will not cause premature machinery component failure.

Over the last ten years, the use of laser alignment and/or alignment computers have become popular. I do not object to their use. In fact, it is encouraged as long as the principles behind proper machinery alignment are understood. Please refer to Figure 13.24.

Accurate alignment results from:

- Knowing the principles involved
- Proper installation of alignment apparatus
- Accounting for alignment apparatus sag (if present)
- Assuring instruments are clean and accurate

We encourage you to review site alignment procedures and alignment apparatus instructions.

Figure 13.24 Accurate alignment results from:

Alignment methods

There are two proper methods of shaft alignment employed in plants around the world:

- Reverse dial indicator method
- Rim and face method

The rim and face method is used only when the reverse dial indicator method cannot be used. At this point, please state where. These facts are presented in Figure 13.25.

Site shaft alignment methods:

- Reverse dial indicator or laser
- Rim and face (where reverse or laser method cannot be used)
- Please state where rim and face alignment is used on site

____

____

Figure 13.25 Site shaft alignment methods

Proper set-up and alignment guidelines

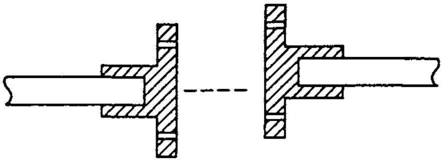
Regardless of the type of alignment utilized, unless alignment apparatus is properly set-up, clean and accurate, the equipment will not be properly aligned.

Today, with the 'modern tools' available (laser, alignment computers, etc.), we have experienced many cases where … 'The aligned equipment wasn't'.

The following information is provided as a 'best practice' guideline to be following regardless of the type of alignment apparatus utilized. Carefully following this information will assure long component life free from the effects of misalignment.

Some general considerations are presented in Figures 13.26 and 13.27 concerning shaft alignment.

Shaft alignment



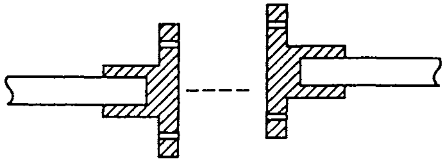
Preliminary considerations

1. Obtain thermal shafts and machine growth calculations establish 'cold offsets'
2. Coupling hubs installed in accordance with OEM's procedures
3. Set proper B.S.E. dimension
4. Test for 'soft foot' 0.002 maximum allowable differential rise
5. Use only stainless shims – minimize number of shims (4 maximum)
6. Shims must straddle hold down bolts

Figure 13.26 Shaft alignment – Preliminary considerations

It is important that proper dimensions be obtained from the equipment manufacturer's instruction book regarding cold offsets and axial separation distance between shaft ends. Many times equipment is presented to the user by the contractor for acceptance when shaft end dimensions are not in accordance with manufacturer's requirements. Proper attention early in the construction phase will alleviate this situation. Coupling hubs must also be installed in accordance with the vendor's procedures. Prior to beginning alignment checks, the base plate machinery interface must be checked for soft foot. This is accomplished by alternately loosening hold down bolts and checking that a maximum differential rise of .002 of an inch is not exceeded. Stainless steel shims should be used only and the number minimized since shims act like a spring. It is the intention to rigidly support equipment. Shims also must straddle hold down bolts to provide maximum surface area and thus minimize high localized stresses on grout and foundation.

Shaft alignment



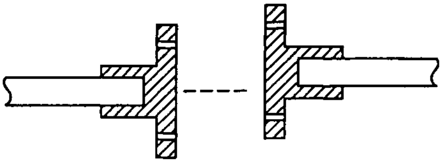
Alignment parameters

1. All piping must be disconnected
2. Reverse dial indicator method shall be used
3. The maximum allowable alignment limits shall be 0.002 T.I.R. per inch of spacer parallel offset
4. Demonstrate results are repeatable
5. Maximum bracket sag = 0.003

Figure 13.27 Shaft alignment – alignment parameters

Piping must always be disconnected for initial alignment. Results should be repeatable and any alignment brackets should be constructed such that their sag is minimized. As previously mentioned, external piping can cause significant stresses in the casing and eventually the bearing brackets, thus affecting equipment internal alignment. It is imperative that alignment change limits be instituted when connecting piping. The maximum allowable movement of the equipment shaft when piping is connected should be limited to 0.002". Figure 13.28 presents other details concerning the proper installation of piping to equipment. Note actual bracket sag must be taken into account during calculations.

Shaft alignment



Alignment change limits when connecting piping

1. Mount dial indicators independent of machinery horizontally and vertically reading on coupling flanges
2. Zero out indicators
3. Tighten using specified sequence and torque values
4. Maximum shaft movement = 0.002
5. Dowell if required by OEM
6. What is the site standard for dowling equipment?

____

____

Figure 13.28 Shaft alignment – Alignment change limits when connecting piping

Both driver and driven equipment should be dowelled in the field when specified by the manufacturer. No dowelling is to be done until all piping and alignment are finalized. Dowels with threads on the top end are preferred to facilitate removal without damage to dowels or machinery.

As previously stated, it has been our experience that many alignment problems came from action taken during the alignment process. Therefore, we have included in Figure 13.29 reverse indicator alignment 'tips'.

Reverse indicator shaft alignment 'tips'

1. Walk unit to be aligned – observe all spring hangers are free
2. Clean all instrument mounting and shim surfaces
3. Confirm indicator bar sag and include in calculations
4. Follow indicator movement with alignment mirror
5. Always rotate shafts in the same direction
6. Rotate shafts at the end opposite you are reading values
7. Be sure SS shims are used
8. Use a maximum of 4 shims
9. Always plot by hand the first time a new 'alignment computer' is used. This will check the computer, your input and confirm 'your understanding of the procedure'.

Figure 13.29 Reverse indicator shaft alignment 'tips'

On the following pages we have included site installation checklists for:

- Steam turbine
- Motor and pump

Installation checklist for turbine

Plant # ________ Equipment # ________ Date ________

Craftsman name and badge # ________

No.	Contents
1.	Base cleaned and prepared, all holes tapped ready for bolts
2.	Check for soft foot and correct distance between coupling hubs ___ Soft foot ___ Turbine Distance between coupling hubs __________ ___ Soft foot ___
3.	Check and log cold alignment figures:
4.	Check and log pipe stress deflection Vertical ________ Horizontal ________ (.002" maximum deflection allowed)
5.	Check and log hot alignment figures:
6.	Check and mark correct oil level on the casing
7.	Check and log coupling float
8.	Grease coupling Check that coupling bolts are torqued to correct settings ________ ft lbs

Approved by: __________
(signature)

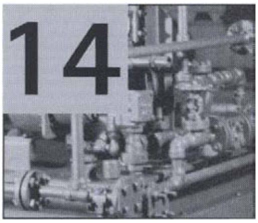
Installation checklist for motor and pump

Plant # ________ Equipment # ________ Date

Craftsman name and badge # ________

No.	Contents
1	Base cleaned and prepared, all holes tapped ready for bolts
2.	Check for soft foot and correct distance between coupling hubs ___ Soft feet ___ ___ Soft feet ___ Motor Distance between coupling hubs Pump ___ Soft feet ___ ___ Soft feet ___
3.	Check and log pipe stress on pump coupling Vertical ________ Horizontal ________ (.002" maximum deflection allowed)
4.	Check and mark correct oil level on the pump casing:
5.	Check and mark the motor magnetic center:
6.	Check and log the clearance between hubs after the spacer is fitted while motor shaft on magnetic center:
7.	Check and log coupling float
8.	Check and log alignment figures
9.	Check that coupling bolts are torqued to correct settings ________ ft lbs

Approved by: ___________
(signature)



Conversion to metric system

Unit Nomenclature

Symbol	Name	Quantity
a	annum (year)	time
bar	bar	pressure
°C	degree Celsius	temperature
°	degree	plane angle
d	day	time
g	gram	mass
h	hour	time
Hz	hertz	frequency
J	joule	work, energy
K	kelvin	temperature
kg	kilogram	mass
litre	litre	volume
m	metre	length
min.	minute	time
N	newton	force
Pa	pascal	pressure
rad	radian	plane angle
s	second	time
t	tonne	mass
V	volt	electric potential
W	watt	power

Conversion tables – Customary to metric (SI) units

Quantity and SI unit	Customary unit	Metric unit	Conversion factor Multiply customary unit by factor to get metric unit	
Space, time				
Length – m	mi	km	1.609 344	E+00
	m	m	1	
	yd	m	9.144	E–01
	ft	m	3.048	E–01
	in	mm	2.54	E+01
	cm	mm	1.0	E+01
	mm	mm	1	
	mil	µm	2.54	E+01
	micron (µ)	µm	1	
Area – m^2	mi^2	km^2	2.589 988	E+00
	yd^2	m^2	8.361 274	E–01
	ft^2	m^2	9.290 304	E–02
	in^2	mm^2	6.451 6	E+02
	cm^2	mm^2	1.0	E+02
	mm^2	mm^2	1	
Volume, capacity – m^3	m^3	m^3	1	
	yd^3	m^3	7.645 549	E–01
	bbl (42 US gal)	m^3	1.589 873	E–01
	ft^3	m^3	2.831 685	E–02
	UK gal	m^3	4.546 092	E–03
	US gal	m^3	3.785 412	E–03
	litre	dm^3	1	
	UK qt	dm^3	1.136 523	E+00
	US qt	dm^3	9.463 529	E–01
	UK pt	dm^3	5.682 609	E–01
	US pt	dm^3	4.731 765	E–01
	US fl oz	cm^3	2.957 353	E+01
	UK fl oz	cm^3	2.841 305	E+01
	in^3	cm^3	1.638 706	E+01
	ml	cm^3	1	

Quantity and SI unit	Customary unit	Metric unit	Conversion factor Multiply customary unit by factor to get metric unit	
Plane angle – rad	rad	rad	1	
	deg (°)	rad	1.745 329	E–02
	min (′)	rad	2.908 882	E–04
	sec (″)	rad	4.848 137	E–06

Conversion to the metric (SI) system

Quantity and SI unit	Customary unit	Metric unit	Conversion factor Multiply customary unit by factor to get metric unit	
Mass, amount of substance				
Mass – kg	UK ton	t	1.016 047	E+00
	US ton	t	9.071 847	E–01
	UK cwt	kg	5.080 234	E+01
	US cwt	kg	4.535 924	E+01
	kg	kg	1	
	lb	kg	4.535 924	E–01
	oz (troy)	g	3.110 348	E+01
	oz (av)	g	2.834 952	E+01
	g	g	1	
	grain	mg	6.479 891	E+01
	mg	mg	1	
	µg	µg	1	
Amount of substance – mol	lb mol	kmol	4.535 924	E–01
	g mol	kmol	1.0	E–03
	std m^3 (0°C, 1atm)	kmol	4.461 58	E–02
	std m^3 (15°C, 1atm)	kmol	4.229 32	E–02
	std ft^3 (60°C, 1 atm)	kmol	1.195 30	E–03

Quantity and SI unit	Customary unit	Metric unit	Conversion factor Multiply customary unit by factor to get metric unit	
Calorific value, heat, entropy, heat capacity				
Calorific value – J/kg (Mass basis)	Btu/lb	MJ/kg	2.326 000	E–03
		kJ/kg	2.326 000	E+00
	cal/g	kJ/kg	4.184	E+00
	cal/lb	J/kg	9.224 141	E+00
Calorific Value –J/mol (Mole basis)	kcal/g mol	kJ/kmol	4.184	E+03
	Btu/lb mol	MJ/kmol	2.326 000	E–03
		kJ/kmol	2.326 000	E+00
Calorific value – J/m^3 (Volume basis – solids and liquids)	therm/UK gal	MJ/m^3	2.320 800	E+04
		kJ/m^3	2.320 800	E+07
	Btu/US gal	MJ/m^3	2.787 163	E–01
		kJ/m^3	2.787 163	E+02
	Btu/UK gal	MJ/m^3	2.320 800	E–01
		kJ/m^3	2.320 800	E+02
	Btu/ft^3	MJ/m^3	3.725 895	E–02
		kJ/m^3	3.725 895	E+01
	kcal/m^3	MJ/m^3	4.184	E–03
		kJ/m^3	4.184	E+00
	cal/ml	MJ/m^3	4.184	E+00
	ft•lb_f/US gal	kJ/m^3	3.581 692	E–01
Calorific value – J/m^3 (Volume basis – gases)	cal/ml	kJ/m^3	4.184	E+03
	kcal/m^3	kJ/m^3	4.184	E+00
	Btu/ft^3	kJ/m^3	3.725 895	E+01
Specific entropy – J/kg•K	Btu/lb•°R	kJ/kg•K	4.186 8	E+00
	cal/g•°K	kJ/kg•K	4.184	E+00
	kcal/kg•°C	kJ/kg•K	4.184	E+00
Specific heat capacity – J/kg•K (Mass basis)	kW·h/kg•°C	kJ/kg•°C	3.6	E+03
	Btu/lb•°F	kJ/kg•°C	4.186 8	E+00
	kcal/kg•°C	kJ/kg•°C	4.184	E+00
Specific heat capacity – J/mol•K (Mole basis)	Btu/lb mol•°F	kJ/kmol•°C	4.186 8	E+00
	cal/g mol•°C	kJ/kmol•°C	4.184	E+00

Quantity and SI unit	Customary unit	Metric unit	Conversion factor Multiply customary unit by factor to get metric unit	
Temperature, pressure, vacuum				
Temperature – K	°R	K	5/9	
(Absolute)	°K	K	1	
Temperature – K	°F	°C	5/9 (°F – 32)	
(Traditional)	°C	°C	1	
Temperature – K	°F	°C	5/9	
(Difference)	°C	°C	1	
Temperature/Length – K/m (Geothermal gradient)	°F per 100 ft	mK/m	1.822 689	E+01
Pressure – Pa	atm	MPa	1.013 250	E–01
		kPa	1.013 250	E+02
	bar	MPa	1.0	E–01
		kPa	1.0	E+02
	at (kg$_f$/cm^2)	MPa	9.806 650	E–02
		kPa	9.806 650	E+01
	lb$_f$/in^2 (psi)	MPa	6.894 757	E–03
		kPa	6.894 757	E+00
	in Hg (60°F)	kPa	3.376 85	E+00
	in H$_2$0 (39.2°F)	kPa	2.490 82	E–01
	in H$_2$0 (60°F)	kPa	2.488 4	E–01
	mm Hg = torr (0°C)	kPa	1.333 224	E–01
	cm H$_2$0 (4°C)	kPa	9.806 38	E–02
	lb$_f$/ft^2 (psf)	kPa	4.788 026	E–02
	μm Hg (0°C)	Pa	1.333 224	E–01
	μbar	Pa	1.0	E–01
	dyn/cm^2	Pa	1.0	E–01
Vacuum, draft – Pa	in Hg (60°F)	kPa	3.376 85	E+00
	in H$_2$0 (39.2°F)	kPa	2.490 82	E–01
	in H$_2$0 (60°F)	kPa	2.488 4	E–01
	mm Hg = torr (0°C)	kPa	1.33 224	E–01
	cm H$_2$0 (4°C)	kPa	9.806 38	E–02
Liquid head – m	ft	m	3.048	E–01
	in	mm	2.54	E+01
Pressure drop/length – Pa/m	psi/ft	kPa/m	2.262 059	E+01
	psi/100 ft	kPa/m	2.262 059	E–01

Quantity and SI unit	Customary unit	Metric unit	Conversion factor Multiply customary unit by factor to get metric unit	
Density, specific volume, concentration, dosage				
Density (gases) – kg/m^3	lb/ft^3	kg/m^3	1.601 846	E+01
		g/m^3	1.601 846	E+04
Density (liquids) – kg/m^3	lb/US gal	kg/dm^3	1.198 264	E–01
	lb/UK gal	kg/dm^3	9.977 644	E–02
	lb/ft^3	kg/dm^3	1.601 846	E–02
	g/cm^3	kg/dm^3	1	
Density (solids) – kg/m^3	lb/ft^3	kg/dm^3	1.601 846	E–02
Specific volume – m^3/kg (gases)	ft^3/lb	m^3/kg	6.242 796	E–02
		m^3/g	6.242 796	E–05
Specific volume – m^3/kg (liquids)	ft^3/lb	dm^3/kg	6.242 796	E+01
	UK gal/lb	dm^3/kg	1.022 241	E+01
	US gal/lb	dm^3/kg	8.345 404	E+00
Specific volume – m^3/mol (mole basis)	litre/g mol	m^3/kmol	1	
	ft^3/lb mol	m^3/kmol	6.242 796	E–02
Concentration – kg/kg (mass/mass)	wt %	kg/kg	1.0	E–02
		g/kg	1.0	E–05
	wt ppm	mg/kg	1	
Concentration – kg/m^3 (mass/volume)	lb/bbl	kg/m^3	2.853 010	E+00
	g/US gal	kg/m^3	2.641 720	E–01
	g/UK gal	kg/m^3	2.199 692	E–01

Quantity and SI unit	Customary unit	Metric unit	Conversion factor Multiply customary unit by factor to get metric unit	
Facility throughput, capacity				
Throughput – kg/s	million lb/yr	t/a	4.535 924	E+02
(mass basis)	UK ton/yr	t/a	1.016 047	E+00
	US ton/yr	t/a	9.071 847	E-01
	UK ton/d	kg/h	1.016 047	E+00
	US ton/d	t/d	9.071 847	E-01
	UK ton/h	t/h	1.016 047	E+00
	US ton/h	t/h	9.071 847	E-01
	lb/h	t/d	4.535 924	E-01
Throughput – m^3/s	bbl/d	t/a	5.803 036	E+01
(volume basis)		m^3/h	6.624 471	E-03
	ft^3/d	m^3/h	1.179 869	E-03
	bbl/h	m^3/h	1.589 873	E-01
	ft^3/h	m^3/h	2.831 685	E-02
	UK gal/h	m^3/h	4.546 092	E-03
	US gal/h	m^3/h	3.785 412	E-03
	UK gal/min	m^3/h	2.727 655	E-01
	US gal/min	m^3/h	2.271 247	E-01
Throughput – mol/s	lb mol/h	kmol/h	4.535 924	E-01
(mole basis)				

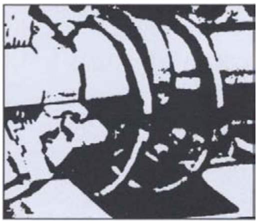
Quantity and SI unit	Customary unit	Metric unit	Conversion factor Multiply customary unit by factor to get metric unit	
Flow rate				
Flow rate – kg/s	UK ton/min	kg/s	1.693 412	E+01
(mass basis)	US ton/min	kg/s	1.511 974	E+01
	UK ton/h	kg/s	2.822 353	E–01
	US ton/h	kg/s	2.519 958	E–01
	UK ton/d	kg/s	1.175 980	E–02
	US ton/d	kg/s	1.049 982	E–02
	million lb/yr	kg/s	5.249 912	E+00
	UK ton/yr	kg/s	3.221 864	E–05
	US ton/yr	kg/s	2.876 664	E–05
	lb/s	kg/s	4.535 924	E–01
	lb/min	kg/s	7.559 873	E–03
	lb/h	kg/s	1.259 979	E–04
Flow rate – m^3/s	bbl/d	dm^3/s	1.840 131	E–03
(volume basis)	ft^3/d	dm^3/s	3.277 413	E–04
	bbl/h	dm^3/s	4.416 314	E–02
	ft^3/h	dm^3/s	7.865 791	E–03
	UK gal/h	dm^3/s	1.262 803	E–03
	US gal/h	dm^3/s	1.051 503	E–03
	UK gal/min	dm^3/s	7.576 820	E–02
	US gal/min	dm^3/s	6.309 020	E–02
	ft^3/min	dm^3/s	4.719 474	E–01
	ft^3/s	dm^3/s	2.831 685	E+01
Flow rate – mol/s	lb mol/s	kmol/s	4.535 924	E–01
(mole basis)	lb mol/h	kmol/s	1.259 979	E–04

Quantity and SI unit	Customary unit	Metric unit	Conversion factor Multiply customary unit by factor to get metric unit	
Energy, work, power				
Energy, work – J	therm	MJ	1.055 056	E+02
		kJ	1.055 056	E+05
		kJ	3.6	E+03
	Btu	kJ	1.055 056	E+00
	kcal	kJ	4.184	E+00
	cal	kJ	4.184	E–03
	ft•lb$_f$	kJ	1.355 818	E–03
	lb•ft	kJ	1.355 818	E–03
	J	kJ	1.0	E–03
	lb•ft^2/s^2	kJ	4.214 011	E–05
	erg	J	1.0	E–07
Power – W	million Btu/h	MW	2.930 711	E–01
	ton of refrigeration	kW	3.516 853	E+00
	Btu/s	kW	1.055 056	E+00
	kW	kW	1	
	hydraulic horsepower–hhp	kW	7.460 43	E–01
	hp (electric)	kW	7.46	E–01
	Btu/min	kW	1.758 427	E–02
	ft•lb$_f$/s	kW	1.355 818	E–03
	kcal/h	W	1.162 222	E+00
	Btu/h	W	2.930 711	E–01
	ft•lb$_f$/min	W	2.259 697	E–02
Power/area – W/m^2	Btu/s•ft^2	kW/m^2	1.135 653	E+01
	Cal/h•cm^2	kW/m^2	1.162 222	E–02
	Btu/h•ft^2	kW/m^2	3.154 591	E–03
Cooling duty – W/W (machinery)	Btu/bhp•h	W/kW	3.930 148	E–01
Specific fuel – kg/J consumption (mass basis)	lb/hp•h	mg/J	1.689 659	E–01
Specific fuel – m^3/J consumption (volume basis)	m^3/kW•h	dm^3/MJ	2.777 778	E+02
	US gal/hp•h	dm^3/MJ	1.410 089	E+00

Quantity and SI unit	Customary unit	Metric unit	Conversion factor Multiply customary unit by factor to get metric unit	
Mechanics				
Velocity (linear) – m/s	mi/h	km/h	1.609 344	E+00
speed	m/s	m/s	1	
	ft/s	m/s	3.048	E–01
	ft/min	m/s	5.08	E–03
	ft/h	mm/s	8.466 667	E–02
	in/s	mm/s	2.54	E+01
	in/min	mm/s	4.233 333	E–01
Corrosion rate – mm/a	in/yr (ipy)	mm/a	2.54	E+01
Rotational frequency – rev/s	rev/s	rev/s	1	
	rev/min	rev/s	1.666 667	E–02
Acceleration – m/s^2	ft/s^2	m/s^2	3.048	E–01
(linear)	gal (cm/s^2)	m/s^2	1.0	E–02
Acceleration – rad/s^2	rad/s^2	rad/s^2	1	
(rotational)				
Momentum – kg•m/s	lb•ft/s	kg•m/s	1.382 550	E–01
Force – N	UK ton$_f$	kN	9.964 016	E+00
	US ton$_f$	kN	8.896 443	E+00
	kg$_f$ (kp)	N	9.806 650	E+00
	lb$_f$	N	4.448 222	E+00
	N	N	1	
Bending moment, – N•m	US ton$_f$•ft	kN•m	2.711 636	E+00
torque	kg$_f$•m	N•m	9.806 650	E+00
	lb$_f$•ft	N•m	1.355 818	E+00
	lb$_f$•in	N•m	1.129 848	E–01
Bending – N•m/m	lb$_f$•ft/in	N•m/m	5.337 866	E+01
moment/length	kg$_f$•m/m	N•m/m	9.806 650	E+00
	lb$_f$•in/in	N•m/m	4.448 222	E+00
Moment of – kg•m^2	lb$_f$•ft^2	kg•m^2	4.214 011	E–02
inertia	in^4	cm^4	4.162 314	E+01

Quantity and SI unit	Customary unit	Metric unit	Conversion factor Multiply customary unit by factor to get metric unit	
Mechanics continued				
Stress – Pa	US ton$_f$/in^2	MPa	1.378 951	E+01
	kg$_f$/mm^2	MPa	9.806 650	E+00
	US ton$_f$/ft^2	MPa	9.576 052	E–02
	lb$_f$/in^2 (psi)	MPa	6.894 757	E–03
	lb$_f$/ft^2 (psf)	kPa	4.788 026	E–02
	dyn/cm^2	Pa	1.0	E–01
Mass/length – kg/m	lb/ft	kg/m	1.488 164	E+00
Transport properties				
Thermal resistance – K•m^2/W	°C•m^2•h/kcal	°C•m^2/kW	8.604 208	E+02
	°F•ft^2•h/Btu	°C•m^2/kW	1.761 102	E+02
Heat flux – W/m^2	Btu/h•ft^2	kW/m^2	3.154 591	E–03
Thermal – W/m•K conductivity	cal/s•cm^2•°C/cm	W/m•°C	4.184	E+02
	Btu/h•ft^2•°F/ft	W/m•°C	1.730 735	E+00
	Kcal/h•m^2•°C/m	W/m•°C	1.162 222	E+00
	Btu/h•ft^2•°F/in	W/m•°C	1.442 279	E–01
	cal/h•cm^2•°C/cm	W/m•°C	1.162 222	E–01
Heat transfer – W/m^2•K coefficient	cal/s•cm^2•°C	kW/m^2•°C	4.184	E+01
	Btu/s•ft^2•°F	kW/m^2•°C	2.044 175	E+01
	cal/h•cm^2•°C	kW/m^2•°C	1.162 222	E–02
	Btu/h•ft^2•°F	kW/m^2•°C	5.678 263	E–03
	Btu/h•ft^2•°R	kW/m^2•K	5.678 263	E–03
	kcal/h•m^2•°C	kW/m^2•°C	1.162 222	E–03
Volumetric heat – W/m^3•K transfer coefficient	Btu/s•ft^3•°F	kW/m^3•°C	6.706 611	E+01
	Btu/h•ft^3•°F	kW/m^3•°C	1.862 947	E–02
Surface tension – N/m	dyn/cm	mN/m	1	
Viscosity – Pa•s (dynamic)	lb$_f$•s/in^2	Pa•s	6.894 757	E+03
	lb$_f$•s/ft^2	Pa•s	4.788 026	E+01
	kg$_f$•s/m^2	Pa•s	9.806 650	E+00
	lb/ft•s	Pa•s	1.488 164	E+00
	dyn•s/cm^2	Pa•s	1.0	E+01
	cP	Pa•s	1.0	E–03
	lb/ft•h	Pa•s	4.133 789	E–04

Quantity and SI unit	Customary unit	Metric unit	Conversion factor Multiply customary unit by factor to get metric unit	
Transport properties continued				
Viscosity – m^2/s (kinematic)	ft^2/s	mm^2/s	9.290 304	E+04
	in^2/s	mm^2/s	6.451 6	E+02
	m^2/h	mm^2/s	2.777 778	E+02
	cm^2/s	mm^2/s	1.0	E+02
	ft^2/h	mm^2/s	2.580 64	E+01
	cSt	mm^2/s	1	



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